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CHAPTER I

Water Turbines—General Characteristics

By P. E. DÉRIAZ

1. Introduction.

In view of the limited extent of this section on water turbines, the authors of Chaps. I–IV have omitted some of the elementary theory of water turbines, on the assumption that the reader will either already be familiar with it or, alternatively, will be able to obtain access to it in any of the standard works. They are thus able to present more information which will be of value to the practical engineer, directly concerned with the development of hydro-electric power.

Chap. I deals with general characteristics of water turbines and contains some information in § 5 on selection of the type of turbine best suited to the conditions of a particular installation. Chaps. II, III and IV cover respectively the principal types of turbines now in use—Pelton, Francis, Kaplan and propeller—in some detail.

It should be noted that certain of the sub-headings, such as the phenomenon of cavitation and the design of the principal component parts of reaction turbines, are treated in more than one of these chapters and that for the fullest information reference should be made to each of them. These chapters deal with the design, manufacture, and testing of water turbines; for the effect which the type and design of turbines have on the planning and layout of power stations as a whole, the reader should refer to Chap. XVI.

2. Classification of Turbines.

Water turbines are subject to more varied conditions than other prime movers. Heads from 5800 ft. down to 3 ft. and single machines with outputs ranging from fractional horse-power to 400,000 b.h.p. are in use at the present date. The design of water turbines has to cover an almost infinite number of combinations of heads, speeds, and outputs; hence it is necessary to classify them according to the types in general

Turbinas de ITAIPU N 1.000.000 HP 715 MW

use, and each type according to the hydraulic conditions for which it gives the best performance.

The water turbines which are best suited for operating electric generators have now almost exclusive consideration from the engineer and belong to four groups:

Pelton wheels, which are impulse turbines.
Francis turbines
Propeller and Kaplan turbines
Dériaz turbines
} which are reaction turbines.

The hydraulic characteristics of any water turbine can be summarized conveniently by reference to its *specific speed*, which is a most useful index to its capabilities, and is represented by the symbol n_s.

The four groups cover a range of n_s as follows:

(*a*) Pelton wheels: from 2 to 8 (British units) for single-jet machines, and up to 11 for double-jet machines.
(*b*) Francis turbines: from 14 to 90.
(*c*) Propeller and Kaplan turbines: from 70 to 220.
(*d*) Dériaz turbines: from 40 to 100 (see Appendix to Chap. III).

Before defining n_s the principle of hydraulic similarity must be considered.

3. Hydraulic Similarity.

Any turbine can be operated under various heads and, at each head, it will run over a range of speeds; for each head, however, there is one speed at which it has the best efficiency. For a well-designed turbine, this speed corresponds to the conditions of smooth entry of water into the revolving runner and to a minimum of losses throughout.

Let H be the head in feet and n the speed in revolutions per minute. All passages by which the water flows through the runner are best suited to the directions of the water velocities prevailing at the designed speed of rotation.

MISMO RODETE
DISTINTO H

* If the same turbine is worked under a different head H', the geometrical picture of the water velocities will be changed, but at a certain speed n' the *direction* of the water velocities in all parts of the turbine will coincide with the direction prevailing under head H at the speed n. Only the magnitude of the velocities has changed. As the total energy available has altered from H to H', the squares of all the velocities and the friction losses must have altered in the ratio of the heads. The velocities have preserved their directions but in magnitude they have changed in the ratio $\sqrt{H'}/\sqrt{H}$.

H ≠ [illegible]
D = [illegible]

At a given point in the turbine and under the head H, assume the absolute water velocity to be C, the water velocity relative to the runner

is W and the peripheral velocity of the runner is U. Under the different head H', there are at the same point the corresponding water velocities C' and W' and the peripheral velocity U'.

Hydraulic similarity prevails when

$$\frac{C}{C'} = \frac{W}{W'} = \frac{U}{U'} = \sqrt{\frac{H}{H'}}$$

C = water velocity
W = " " relative to the runner
U = velocity of the runner

The speeds of revolution n and n' and the discharges Q and Q' are in the same ratio as the velocities and, therefore,

$$\frac{n}{n'} = \frac{Q}{Q'} = \sqrt{\frac{H}{H'}}$$

n = speed
Q = discharge
H = head

The outputs N and N' developed at the turbine shaft, being the product of head and discharge, are in the ratio

N = output

$$\frac{N}{N'} = \frac{HQ}{H'Q'} = \frac{H\sqrt{H}}{H'\sqrt{H'}}$$

Finally, if η_h and η_h' are the hydraulic efficiencies of the turbine under the operating conditions just defined, then $\eta_h = \eta_h'$ because of congruent angles and similarity of losses.

** If two turbines are considered to be geometrically similar and have runners of diameters D and D^* respectively, corresponding dimensions are in the ratio

DISTINTO RODETE

$D \neq D^*$

$$\frac{D}{D^*}$$

MISMO H

$H = H^*$

If we assume that these two turbines operate under <u>*identical heads*</u>, at a certain pair of speeds n and n^*, the corresponding velocities C and C^*, W and W^*, U and U^* form identical patterns in *direction and <u>magnitude</u>*. Because

$$U = \frac{\pi D n}{60} = U^* = \frac{\pi D^* n^*}{60}$$

the speeds of revolution are in *inverse* ratio of the diameters:

$$\frac{n}{n^*} = \frac{D^*}{D}$$

The discharges are proportional to the areas of the passages and, therefore, in the ratio

$$\frac{Q}{Q^*} = \left(\frac{D}{D^*}\right)^2$$

The horsepowers N and N^* are in the ratio

$$\frac{N}{N^*} = \frac{QH}{Q^*H} = \left(\frac{D}{D^*}\right)^2$$

and, because of congruent velocities, the efficiencies are identical except for the friction losses which become proportionately smaller for larger diameters (see " Scale Effect ", § 8).

4. Specific Speed.

Specific speed is defined as *the speed in revolutions per minute* at which a turbine would run at the best efficiency for full guide-vane opening under a *head of one foot*, its dimensions having been adjusted to produce *one horse-power*.

Consider a turbine of runner diameter D operating under a head H (both in feet) and giving at full guide-vane opening the output N in b.h.p. at the best efficiency and at a speed n measured in r.p.m.

From § 3 it is clear that, by reducing the head to *one foot*, the same turbine will operate at its best efficiency when its speed of revolution becomes

$$n_1 = \frac{n}{\sqrt{H}} = \text{revolutions per minute under 1 ft. head;}$$

$$\frac{Q}{Q'} = \frac{n}{n'} = \sqrt{\frac{H}{H'}}$$

1 ft

the discharge becomes $Q_1 = \dfrac{Q}{\sqrt{H}}$; and

$$N_1 = \frac{N}{H\sqrt{H}} \text{ is the output under 1 ft. head.}$$

$$\frac{N}{N'} = \frac{H\sqrt{H}}{H'\sqrt{H'}}$$

In order to reduce this power N_1 to *one* h.p., the diameter D of the runner is now altered to D^*, so that

$$\frac{N_1}{1} = \left(\frac{D}{D^*}\right)^2 \quad \text{or} \quad \frac{D}{D^*} = \sqrt{N_1}$$

By this change of diameter, the speed is altered in inverse ratio

$$\frac{n^*}{n_1} = \frac{D}{D^*} = \sqrt{N_1}$$

and the n^* obtained is the specific speed n_s by definition. Therefore

$$n_s = n_1\sqrt{N_1} = \frac{n}{\sqrt{H}}\sqrt{\left(\frac{N}{H\sqrt{H}}\right)}$$

$$n_s = n\frac{N^{1/2}}{H^{5/4}}$$

HP

RPM

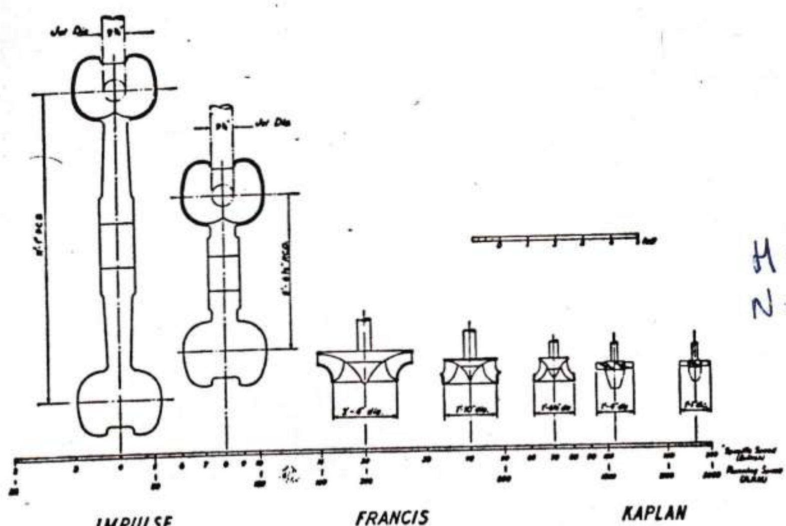
ft
m

or, as usually written,

$$n_s = \frac{n}{H}\sqrt{\left(\frac{N}{\sqrt{H}}\right)} \quad \text{or} \quad n\frac{\sqrt{N}}{H^{5/4}} \quad \text{or} \quad n\frac{N^{1/2}}{H^{5/4}}$$

If *H is in feet* and *N in British horse-power*, n_s is obtained in British units. Should metric units be used, *H in metres* and *N in metric horse-power*, the result is n_s (metric). For conversion, the following relation applies:

$$n_s \text{ (metric)} = 4{\cdot}45\ n_s \text{ (British)}$$



H = 40 ft
N = 100 bhp } ct

Fig. 1.1.—Classification of water turbines according to specific speed n_s

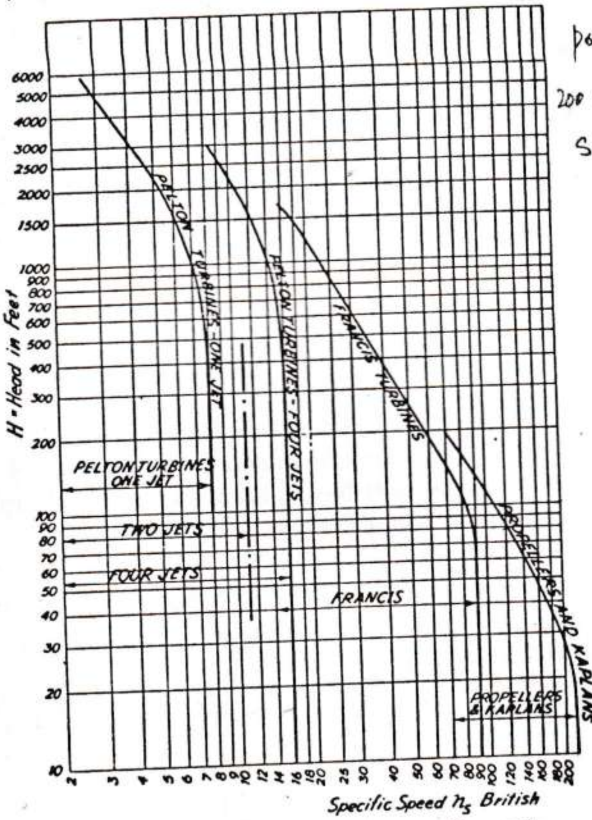
The comparative size of the runners can be gauged from the dimensions and the scale. Each runner develops 100 b.h.p. under 40 ft. net head.
The normal running speeds are equal to 10 times n_s (British).

Fig. 1.1 shows three types of turbine runners: Pelton, Francis, and Kaplan, placed in their respective positions on a baseline graduated in specific speeds (British). These runners are all drawn to the same scale, and the relative sizes are such that each would produce the same output under the same head. For practical reasons, the output has been taken as 100 b.h.p. for a head of 40 ft. The running speed in r.p.m. works out to $10n_s$ as noted in the scale at the bottom of the figure. Fig. 1.1 is intended to convey an appreciation of the very wide range of shapes, sizes and speeds that the turbine designer can offer. Between the seven runners drawn, intermediate sizes and transitions in proportions can be visualized.

For each individual hydro-electric scheme the most suitable type of turbine must be decided upon. This subject is treated in § 5.

5. Choice of Type of Turbine.

The head H in feet under which the turbine will operate gives the first guide to the selection of the type of turbine. Referring to fig. 1.2,



por encima de 2000' P ton
200 < H < 2000' Pelton y ancis
siendo el rango de
600' a 200' mas p a Franc
200 > H > 80' Fra cis y
He ice o K
H < 80' Helice o Kap.
NO SE EXCLUY EL
USO DE PE TON P
E) PARA H= 100' pe
es muy raro

Fig. 1.2.—Limits of head for various specific speeds

The field of operation for each type ot turbine is limited by its highest and lowest n_s, and by the maximum head under which it can safely be operated.

the field of operation of each type is shown in relation to the head H and specific speed n_s. By following the horizontal line corresponding to the given head, the types that can be used effectively can be seen at a glance.

As an example, for a head in excess of 2000 ft., the Pelton turbine

offers the only solution. For a head of 400 ft., the Pelton and Francis types can both be used. Again, should the head be 100 ft., the horizontal line crosses the field of all the three types of turbine. Any one of the three could be used so that a further selection according to n_s becomes necessary.

The total horse-power to be installed must be known and the number of machines then chosen by economic considerations of load factor, extent of water storage if any, cost of power house, convenience of operation and maintenance. Once the output per machine has been decided, information must be obtained concerning the suitable speeds for which the generator can be constructed economically. From these data:

N = turbine output in horse-power at full load,
H = the effective head in feet,
n = normal running speed in r.p.m.,

the specific speed is calculated as

$$n_s = \frac{n}{H}\sqrt{\left(\frac{N}{\sqrt{H}}\right)}$$

The appropriate specific speed is indicated in fig. 1.2, which shows the upper limit of head for which each particular n_s is suitable for each type of turbine. Should the head exceed this permissible limit, a lower speed n or a lower output N must be adopted in order to reduce the value of n_s.

The range of specific speeds will be noted:

Pelton turbines with *one* jet extend up to $n_s = 8$ (British).
Pelton turbines with *two* jets extend up to $8\sqrt{2} = n_s = 11$ (British).
Pelton turbines with *four* jets extend up to $8\sqrt{4} = n_s = 16$ (British).
The Francis turbine range from $n_s = 14$ to $n_s = 90$ *overlaps* the Pelton range (four jets) and the propeller and Kaplan range with n_s over 70.

Since the speed of the generator can generally be selected for several suitable numbers of pole pairs, the appropriate specific speed is not limited to one value only. The overlap is thus considerably extended and, in many cases, the problem arises of selecting from two types of turbine, either of which could be used.

Here a wider knowledge of the advantages and disadvantages of each type will assist, especially with respect to efficiency when running at part load. This is shown in fig. 1.3, where efficiency relative to output is illustrated for some typical examples.

Should the machines be called upon to operate for long periods at

small loads, the Pelton turbine would have preference over the Francis turbine. Similarly, if the choice lay between two Francis turbines, that with the lower specific speed would be the more suitable, whereas, if the choice lay between a Kaplan turbine and a propeller turbine or a Francis turbine with a high specific speed, the Kaplan type would be preferred. This is because the flattest efficiency curve is obtained from the Kaplan turbine, followed by the Pelton turbine, the low-specific-speed Francis turbine, the higher-specific-speed Francis turbine, and finally the propeller turbine which has the most peaked form of efficiency curve.

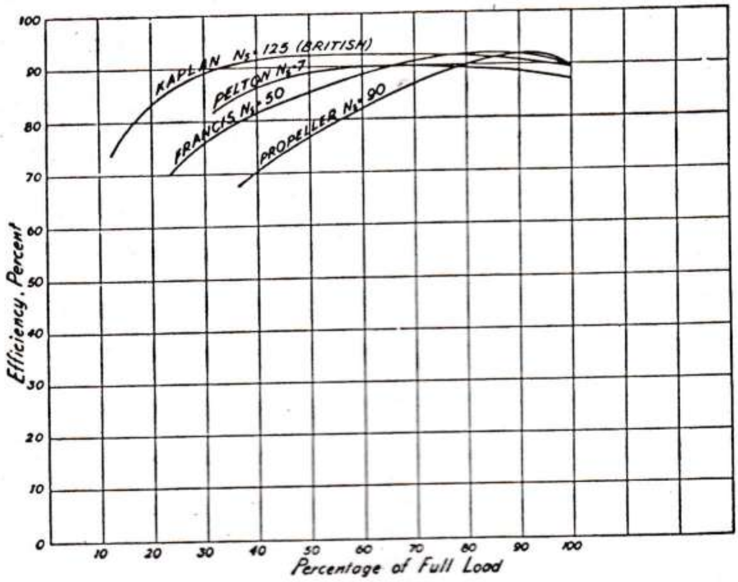


Fig. 1.3.—Efficiencies at various loads

The efficiencies obtainable at partial guide-vane opening differ considerably according to the specific speed n_s.

Generally, a cheaper machine and a less costly power house result from a higher speed of revolution.

Fig. 1.2 requires some further explanation. The upper limits for the heads given are obtained by computation from existing plants. They are not to be regarded as absolute and final but indicate that turbines have been built for and are in successful operation at heads below this line. The limit is, therefore, subject to revision as further experience with new plant becomes available.

It must not be assumed that the highest possible specific speed is

always desirable, and the choice of a value of n_s below the limit line does not mean that the plant is not modern. Again, if the value of n_s is chosen on the limit or very near to it, it is advisable to inquire into the behaviour of existing similar machines, because the limit line is no guarantee of perfect performance; experience concerning newly installed plant is not readily available until some years of operation have passed.

Apart from statistical information from existing installations, the limitations to n_s fall under various headings:

(a) *Hydraulic Limitations.*

The curve shows a limit of $n_s = 8$ for single-jet Pelton turbines (also $n_s = 11$ for two jets, $n_s = 16$ for four jets). This limit is due to space not permitting a larger jet diameter in relation to the pitch-circle diameter of the runner. This ratio of jet diameter to runner (pitch-circle) diameter fixes the value of n_s.

(b) *Mechanical Strength.*

In § 9 it is shown that any one type of turbine of given n_s, characterized by all its geometrical proportions, is subjected to stresses that are proportional to the head under which it operates. In deciding on the use of a given n_s for a head higher than that of the prototype, the designer faces new problems of mechanical strength (and bearing-surface pressures). Sometimes by changing the materials used, higher stresses can be allowed, but more often the proportions of the turbine must be modified, for instance by increasing the thicknesses. This causes some departure from true geometrical similarity and may affect performance.

(c) *Best Maximum Efficiency.*

Experience shows that each type of turbine, Pelton, Francis, propeller and Kaplan, has a best maximum efficiency over a narrower field than generally shown in fig. 1.3. For instance, impulse wheels may give best efficiencies at very high heads that may be inferior to those at lower heads. The very large values of n_s attainable with Kaplan and propeller turbines depend on high recovery in the draft tube of the velocity head, and extremely high specific speeds are obtainable only at some sacrifice of efficiency. It is not to be inferred that this is bad practice; there are notable instances of flood conditions where maximum output at low heads is of paramount importance. In such cases the Kaplan turbine with its generous overload capacity is at its best, and it does not matter greatly if the efficiency percentage drops a little, so long as the required output is available.

(d) *Reynolds' Number.*

Reynolds' number provides a basis for comparing machines, according to the nature of the flow through them, when they are geometrically similar but are of different sizes, as defined by the diameter D, and operate under various heads which affect the velocity v of the water at any given point.

The Reynolds number is

$$R = \frac{vD}{\nu}$$

$R = \frac{VD}{\gamma}$ γ = kinematic viscosity
V = velocity of the water
D = diameter

where ν is the kinematic viscosity of water, which can be considered constant because variations in temperature are small. Since v is proportional to $\sqrt{H}$, one and the same runner operates at increasing Reynolds numbers R as the head increases, maintaining hydraulic similarity.

Since no information from actual plant exists beyond the limits given in fig. 1.2, there is no assurance that the nature of eddy formation will remain acceptable above such heads. In other words, there is a possibility of unexpected vibrations if these limits of head are exceeded. The size of the unit has a similar effect. For one and the same head, turbines of identical n_s but of larger diameters than any actually in operation may be required to work at higher Reynolds numbers and may then show unexpected vibrations because of eddies. The information on this aspect available at present is very inadequate. Statistical methods are difficult to apply because details of design vary widely and are seldom divulged. Therefore, it is wise to be cautious and conservative. The influence of an increasing Reynolds number is found in the *scale effect* (§ 8).

(*e*) *Setting*.

As mentioned in § 6, in order to ensure satisfactory operation, reaction turbines of given n_s require a lower setting of the turbine in relation to the tailwater level when the head H increases. This may result in such unusually deep excavation that the cost of it may well absorb any saving on the machines arising from a higher speed. Also, vibrations at part load due to the phenomena shown in fig. 3.23 become more severe due to the higher energy rejected into the draft tube.

(*f*) *Generator*.

For very large outputs, the generator designers, having particular regard to safety at maximum runaway speed, may fix a limit to the permissible operating speed. The ratio of maximum runaway speed to optimum running speed n is ascertained from model turbines tested in the laboratory, and generally this ratio increases with the specific speed n_s.

6. Turbine Setting and Cavitation.

Impulse turbines, i.e. Pelton wheels, must operate well clear of the tailwater to ensure proper ventilation. Should the tailrace water be subject to large variations in level, the Pelton turbine must be placed sufficiently high to clear it at all times. An appreciable head may thus be lost permanently, and this is of considerable importance if the tailwater level should be subject to high floods or tides.

In such cases, preference is given to the Francis turbine, where the reverse prevails, i.e. the runner must be placed *lower* than a certain permissible level relative to the minimum tailwater level. This applies to all reaction turbines. If they are placed too high, their operation is affected by the phenomenon of cavitation, resulting in pitting and erosion of the runner blades to various degrees, and also noise and vibrations. In the most severe cases this may result in gradual destruction by fatigue due to repeated hammering.

Each type of reaction turbine has its own susceptibility to cavit-

ation and the following considerations will define the limits allowable for the static suction H_s (fig. 1.4).

H_s represents the elevation in feet of the runner above the minimum tailwater level for one turbine operating at full load. The velocity of water at discharge from the runner is C_2 and that at the exit of the draft tube C_4, both in feet per second. Assuming purely axial discharge

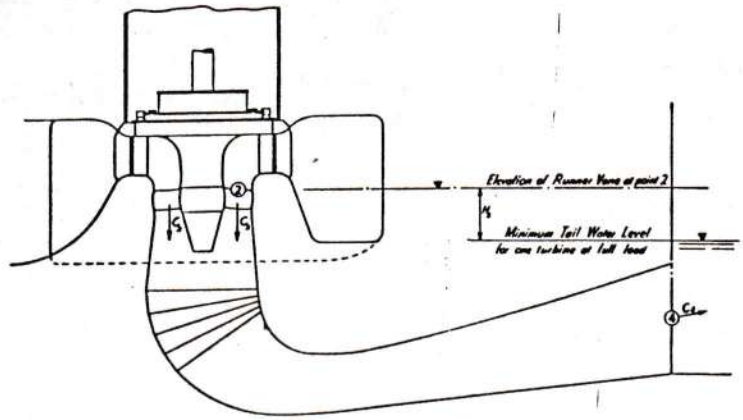


Fig. 1.4.—Setting of a reaction turbine

In order to avoid cavitation, the elevation H_s of the runner above minimum tailwater level must not be allowed to exceed

$$H_s = H_B - \sigma H.$$

Values of σ for each particular n_s are given in fig. 1.5.

H_B is obtained from fig. 1.6.

and even distribution of velocity at entrance and exit of draft tube, the velocities C_2 and C_4 are in inverse ratio to the cross-sectional areas of the water passages at the corresponding places.

The equation of energy for these two sections is

$$\frac{p_2}{\gamma} + \frac{KW_2^2}{2g} + \frac{C_2^2}{2g} + H_s = \frac{C_4^2}{2g} + \sum_2^4 h_z + H_a \quad . \quad . \quad (1)$$

where $\sum_2^4 h_z$ is the sum of losses from section 2 to section 4 in feet head of water,

H_a is the atmospheric pressure over the tailrace in feet of water,

p_2/γ is the absolute pressure at point 2, in feet of water,

$\frac{KW_2^2}{2g}$ is the local depression at the back of the runner vanes, caused by the distribution of velocity W_2 peculiar to each type of turbine. K is a suitable coefficient.

Cavitation is avoided as long as the absolute pressure p_2/γ remains well above the vapour pressure of the water. If the height of the barometric water column in feet is H_B, the condition for cavitation-free operation is $p_2/\gamma > H_a - H_B$ which can be written from (1)

$$\frac{p_2}{\gamma} = H_a + \frac{C_4^2}{2g} + \sum_2^4 h_z - \frac{KW_2^2}{2g} - \frac{C_2^2}{2g} - H_s > H_a - H_B = H_v$$

If η_D is the recovery factor of the velocity head at runner exit, then

$$\eta_D \frac{C_2^2}{2g} = \frac{C_2^2}{2g} - \frac{C_4^2}{2g} - \sum_2^4 h_z$$

and the previous inequality becomes

$$H_B - H_s > \eta_D \frac{C_2^2}{2g} + \frac{KW_2^2}{2g}$$

Supposing it is observed from the behaviour of a particular turbine that cavitation begins at a certain critical value of suction head H_s which will be designated by H_c, it is proposed to calculate the corresponding critical value H_c' when the same turbine operates under a different head H'. The comparison must be based on conditions of hydraulic similarity. Therefore, all velocity heads and friction losses are in proportion to the operating head H. By dividing both sides of the inequality by H, then

$$\frac{H_B - H_c}{H} = \eta_D \frac{C_2^2}{2gH} + \frac{KW_2^2}{2gH}$$

Because of similarity of hydraulic conditions,

$$\frac{C_2^2}{H} = \frac{C_2'^2}{H'} \quad \text{and} \quad \frac{W_2^2}{H} = \frac{W_2'^2}{H'}, \quad \eta_D = \eta_D', \quad K = K'.$$

The right side of the equation $\eta_D\, C_2^2/(2gH) + KW_2^2/(2gH)$ remains the same in changing from the head H to the head H'. It can therefore be considered as a measure of the susceptibility of this particular turbine to cavitation, and be called its " critical sigma " σ_c, a coefficient independent of the head. Observations of the critical suction H_c under the head H permit determination of it as

$$\sigma_c = \frac{H_B - H_c}{H}$$

$H_B = H_a - H_v$ (presion atmosferico - presion de vapor)

H_c = critical suction head

H = total head

and again for the head H' and the barometric pressure H_B',

$$\sigma_c = \frac{H_B' - H_c'}{H'}$$

from which

$$H_c' = H_B' - \sigma_c H'$$

This particular turbine will be free from cavitation as long as the static suction H_s' remains less than the critical static suction H_c'. Therefore, for any head it is necessary to ensure that

$$H_s < H_B - \sigma_c H$$

$H_S < H_B - \sigma_c$
CONDICION P
NO CAVITAC

The larger the susceptibility coefficient σ_c, the more readily will the turbine cavitate and, to counter this, the lower must the turbine be set in relation to the tailwater level; σ_c is a pure number which characterizes each particular turbine. Any turbine that is geometrically similar and operates under similar hydraulic conditions has the same value of σ_c. With a different design or proportions σ_c will generally be different. As can be expected, the specific speed n_s has a great influence on the critical sigma, as is seen from the following considerations:

As given in Chap. III, § 3, the exit diameter D_2 of all reaction turbines is related to the speed and discharge by

$$\frac{D_2^3 n}{Q_2} \approx \text{constant for all specific speeds}$$

From this it is possible to calculate

$$\eta_D \frac{C_2^2}{2gH} \approx 3{\cdot}6 \times 10^{-4}\,(n_s)^{4/3} = \text{dynamic suction}$$

Fig. 1.5 shows this function of n_s. From experimental data, safe values of $KW_2^2/(2gH)$ follow the law

$$K\frac{W_2^2}{2gH} \approx 4{\cdot}0 \times 10^{-6}\,(n_s)^{7/3} = \text{depression at back of vane}$$

Values of this quantity have been shown in fig. 1.5 in order to give the curve of permissible σ. It can be seen how rapidly σ increases with n_s, and thus correspondingly its importance at large specific speeds.

It must always be remembered that the coefficient K depends to a great extent upon each particular design of turbine, namely, the specific loading on the runner blades, their curvatures, and the ratio of chord to pitch.

Some information on this aspect, as related to Francis-runner design, will be found in Chap. III, § 3.

For turbines already installed, from which data concerning cavitation behaviour are available, the actual values of the suction head are recorded. The value

$$\sigma = \frac{H_B - H_s}{H}$$

is then calculated and can be plotted against n_s as in fig. 1.5. It will be seen that the curve of σ given there represents actual installation values for the lowest tailwater levels with some small margin of safety over the " critical sigma " σ_c.

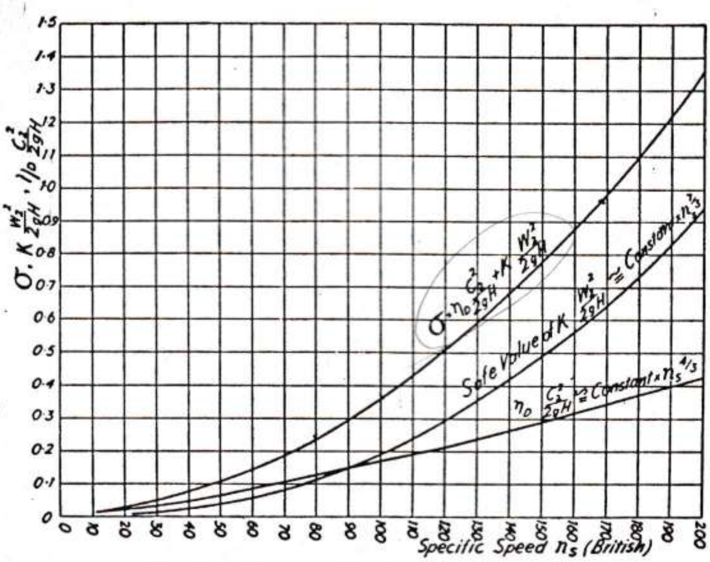


Fig. 1.5.—*Sigma* as a function of *specific speed*

The values of σ shown correspond to safe operation for well-designed turbines.

The experimental determination of σ_c is carried out with small scale-model turbines in the laboratory. The point at which cavitation starts is determined from visual observation of the runner and by the discontinuity that occurs in the performance under conditions of hydraulic similarity when related to the value of $(H_B - H_s)/H$, which is varied by altering either H_s or H or both. By plotting the efficiency, discharge and output under unit head at constant unit speed, against $(H_B - H_s)/H$, the point of discontinuity is made apparent. This

determines the value of the critical sigma σ_c. In order to facilitate the calculation of the critical suction head H_c for each particular installation, fig. 1.6 is given where the height of the water barometer in feet is plotted against the altitude in feet above sea-level, and, for various temperatures of water, H_B can be read off directly as a function of these two variables.

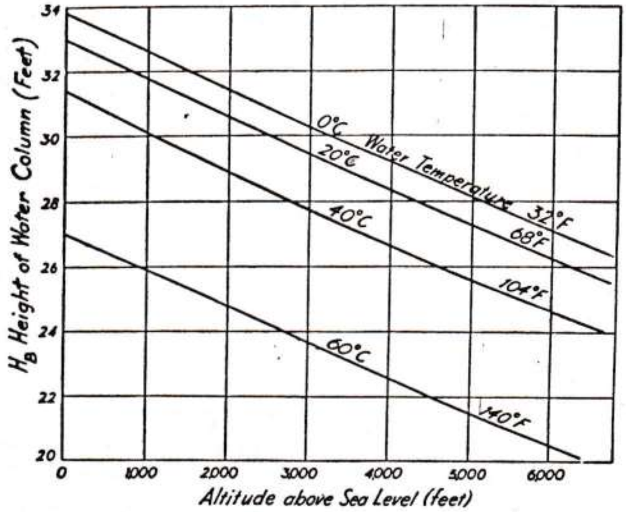


Fig. 1.6.—The height of barometric water column H_B depends upon the atmospheric pressure H_a and the temperature. It can be read off with sufficient accuracy in terms of the altitude above sea-level and the temperature of the water.

7. Model Tests.

Test data that can be obtained from field-testing of turbines are of necessity very restricted. Extensive tests on large installations are costly and usually the time available is very short, because the plant is required in service at the earliest possible date for financial and operational reasons. There are also hydraulic limitations: the head may not be varied, the speed generally is constant and the load available may not be as steady as is desirable for accurate observations.

In order to obtain complete information regarding the characteristics and performance of a water turbine, it is usual to carry out full laboratory tests. A complete replica at a reduced scale is run under a small head. The head is not reduced for the purpose of maintaining the scale, but in order to keep down the rotational speed and the output. As the head is produced artificially by pumping, the consumption of power is important in the economics of the laboratory.

The tests are applied to all types of turbines. Because of their wide disparity, it is essential to use a higher head for tests of impulse turbines, as otherwise the proportions become impracticable. It is generally desirable to arrange for test turbines to be of such dimensions that the turbine output is not less than 5 b.h.p. or more than, say, 50 b.h.p. Too small an output may necessitate excessively fine technique for measurements.

In order to preserve the exact geometrical similitude to the full-size machines, too small a scale must be avoided. This would lead to stringent requirements for accuracy in production of the model and would demand such specialized workmanship as could not be obtained with the usual production standards of the turbine makers.

Because of these considerations, a model-turbine size is chosen where the jet diameter for impulse runners is 1–2 in. at normal full load; while for reaction turbines the runner diameter may be 1–2 ft.

As mentioned, the head is provided by a motor-driven pump; the discharge is measured, either by a calibrated weir or by the volumetric method. The output is absorbed in a Prony brake, which is found the best for work at extremely varied speeds and torques.

The tests consist in running the turbine for a given position of the spear or guide-vane apparatus from standstill to maximum runaway, and measuring for each speed, when steady conditions prevail, the data from which the efficiency is calculated. The test figures of discharge and speed are reduced to unit head and the results plotted in curves of efficiency against speed and discharge against speed.

After covering the complete range of spear or guide-vane openings, the general characteristic curves are worked out. Examples for Pelton, Francis, propeller, and Kaplan turbines respectively are illustrated in figs. 1.7–1.10.

The abscissae are the speeds in r.p.m. reduced to unit head,

$$n_1 = \frac{n}{\sqrt{H}}$$

The ordinates are the discharges reduced to unit head,

$$Q_1 = \frac{Q}{\sqrt{H}}$$

For each guide-vane opening (or spear opening), a curve gives the discharge Q_1 versus speed n_1. The curves of efficiencies are then drawn by joining together the points of equal efficiency over the whole range of discharges. These general characteristic curves are sometimes called

" niveau curves " from the resemblance of the lines of constant efficiency to contour lines on a map; a more common term is " mussel curves " because of their likeness to the lines of growth of the shell-fish of that name.

For turbines intended to drive synchronous generators at a nearly constant frequency, a fixed speed of revolution is required and this is

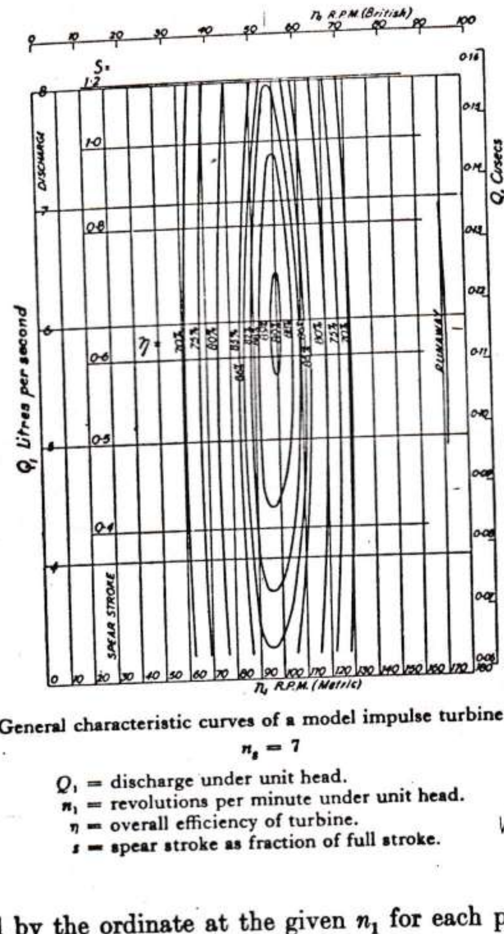


Fig. 1.7.—General characteristic curves of a model impulse turbine (Pelton wheel)

$n_s = 7$

Q_1 = discharge under unit head.
n_1 = revolutions per minute under unit head.
η = overall efficiency of turbine.
s = spear stroke as fraction of full stroke.

$n_1 = n_{11} = \frac{nD}{\sqrt{H}}$

Ver 1.10

represented by the ordinate at the given n_1 for each prevailing head. A change in head is reflected by a change in n_1, and therefore for one and the same speed of revolution different heads are represented by different ordinates.

The field of all the discharges versus speeds that can occur is limited on the right-hand side by the runaway curve, which is the line of zero

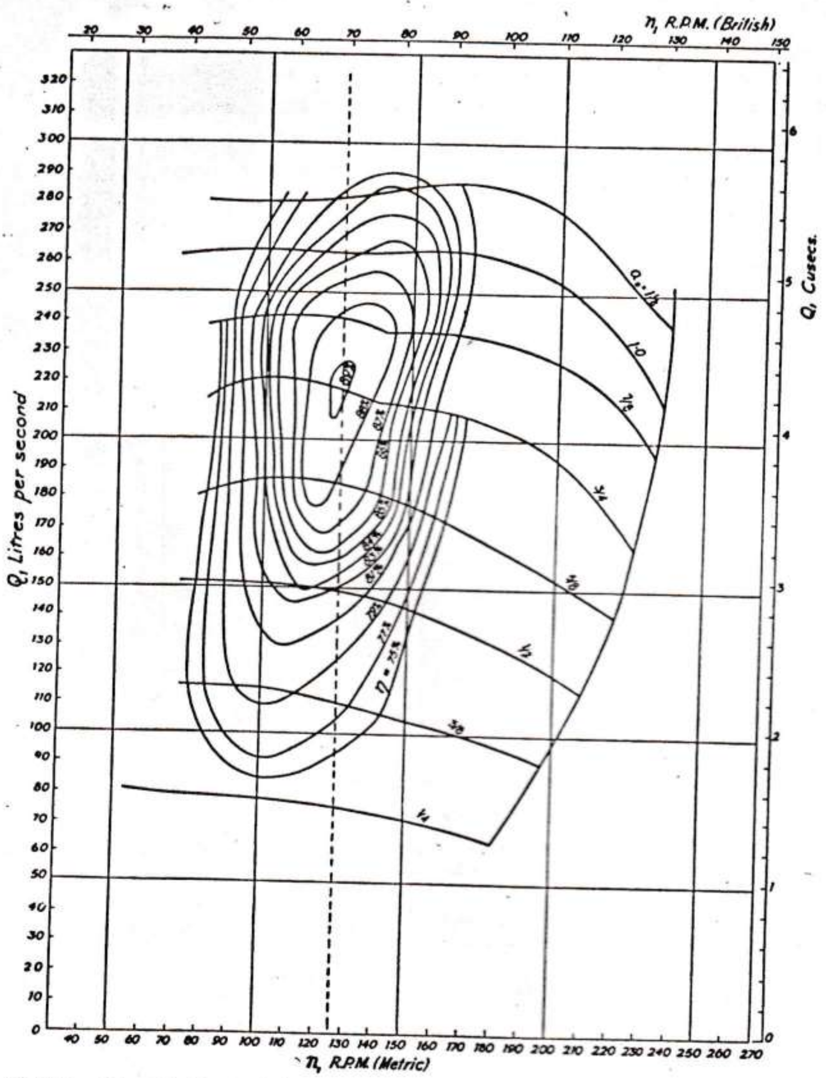


Fig. 1.8.—General characteristic curves of a model Francis turbine of medium specific speed $n_s = 50$

Q_1 = discharge under unit head.

Ver 1.10

n_1 = revolutions per minute under unit head.

η = overall efficiency of model turbine, replica of the full-size turbine from spiral casing inlet to draft tube outlet.

a = gate apparatus opening as fraction of full opening.

efficiency. This runaway of the model test turbine takes place when the turbine is freed from all external load. For designing the generator, the runaway which is of interest is that of the whole rotating element, turbine runner and generator rotor. This is lower than the runaway of turbine alone, because of the considerable windage and other friction losses of the generator at this abnormally high speed.

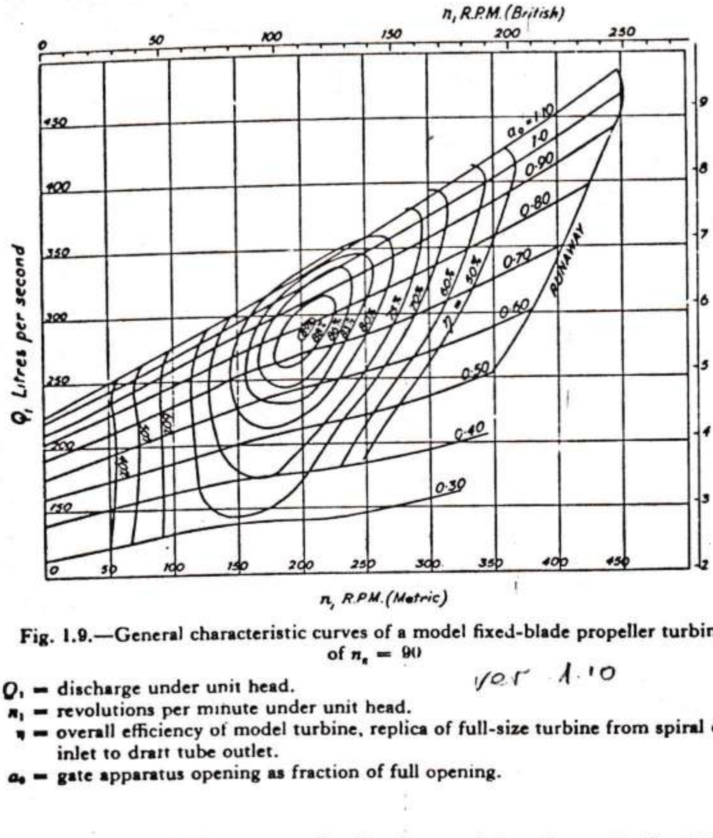


Fig. 1.9.—General characteristic curves of a model fixed-blade propeller turbine of n_s = 90

yor 1.10

Q_1 = discharge under unit head.
n_1 = revolutions per minute under unit head.
η = overall efficiency of model turbine, replica of full-size turbine from spiral casing inlet to draft tube outlet.
a_0 = gate apparatus opening as fraction of full opening.

The characteristic curves of a Kaplan turbine shown in fig. 1.10 are compiled from a series of curves (fig. 1.9) each corresponding to a different runner-blade opening. The curves of equal efficiencies here are the envelopes of the efficiency curves for each runner-opening superimposed, and are valid only for the optimum combination of runner-blade/guide-vane openings.

The absolute maximum runaway curve arises from a combination which differs from that suitable for maximum efficiency. In difficult

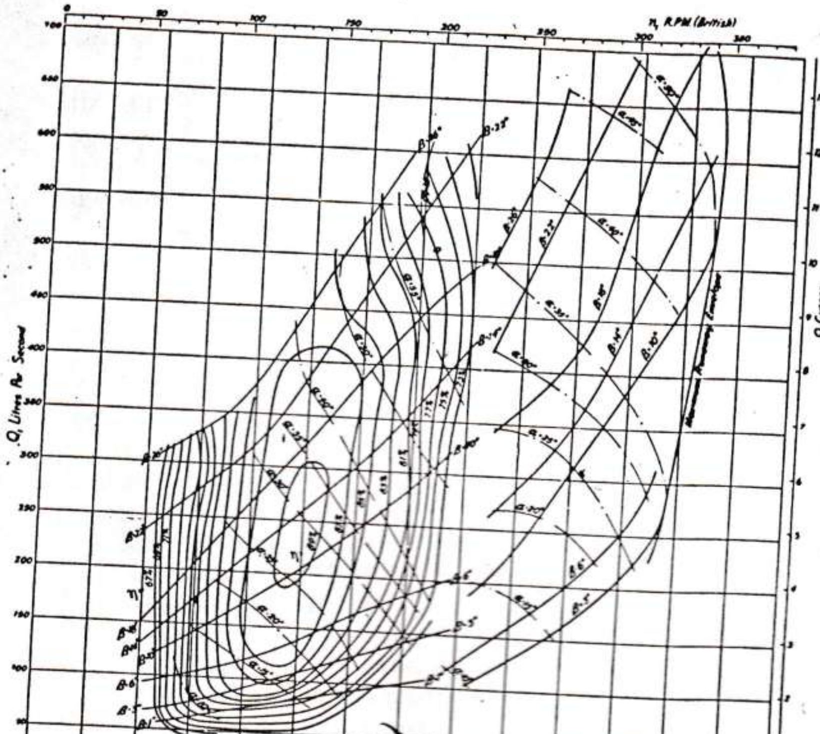


Fig. 1.10.—General characteristic curves of a model Kaplan turbine of $n_s = 125$

Q_1 = discharge under unit head.

n_1 = revolutions per minute under unit head. n_{11} (velocidad unitaria)

η = overall efficiency of model turbine, replica of full-size turbine from spiral casing inlet to draft tube outlet.

α = Guide-vane opening angle in degrees.

β = Runner-vane opening angle in degrees.

The efficiency curves shown correspond to the best combination of guide-vane and runner-vane openings.

cases, judicious limitation of maximum guide-vane openings can reduce the runaway speed very substantially, and thus lighten the task of the generator designer.

8. Scale Effect.

The model test turbines are scaled-down replicas of the full-size machines. The test results from the model turbine are used to ascertain

the anticipated full-size turbine efficiency when working under conditions of strict hydraulic similarity.

As a first approximation, the efficiencies may be considered identical, the losses in each case being proportional to the velocity heads. It has, however, been recognized for many years that, with increase in size, reaction turbines give efficiencies greater than those of the model. This is due to the scale effect.

$\eta_M < \eta_P$

As long as the full-size turbine had a diameter no more than two or three times that of the model, the scale effect gave to the turbine builder just the margin which he wanted over the efficiency he had guaranteed. Furthermore, water measurements in the field are with reason considered subject to greater inaccuracies than in the laboratory. The tolerance of 2 per cent in the guaranteed efficiency was then of the same order as the scale effect.

With further increase in size, however. the ratio between full-size and model diameters became so large as to make the scale effect of importance in comparing competitive offers from various turbine builders. It therefore became necessary to be able to forecast with sufficient accuracy, by taking into account the scale effect, the efficiencies obtainable in the field.

As the field test, which is seldom accurate within more than one per cent, provides the only experimental basis for determination of the scale effect relative to the model turbine test, the calculation of scale effect is a controversial subject.

If η = the efficiency of the prototype turbine,
η' = the efficiency of the model turbine which is geometrically similar to it in every dimension, inclusive of the clearances at the water seals and the relative roughness of all surfaces, ?
impossible
D, D' = the respective diameters,
H, H' = the respective heads,

the losses are $1 - \eta$ and $1 - \eta'$ respectively. The problem is to determine the ratio of losses

$$\frac{1-\eta}{1-\eta'}$$

from which the efficiency η could be calculated, knowing the efficiency η' of the model.

One of a number of solutions offered by Moody is to assume the ratio

of losses to be dependent solely on the diameter ratio

$$\frac{D'}{D} \text{ and of the form } \frac{1-\eta}{1-\eta'} = \left(\frac{D'}{D}\right)^n$$

The exponent n was then derived from a number of field and model data and was found to be $n = 0{\cdot}25$; but $n = 0{\cdot}20$ is recommended by the authors as the more conservative figure.

$$\frac{1-\eta}{1-\eta'} = \left(\frac{D'}{D}\right)^{1/5} = \text{Moody scale effect.}$$

Since, however, the friction losses in pipes are known to depend on the Reynolds number, it is better to consider by analogy that the friction losses in the turbine runner are a function of the Reynolds numbers

$$R = \frac{Dv}{\nu} \quad \text{and} \quad R' = \frac{D'v'}{\nu'}$$

for the model turbine.

By assuming the same kinematic viscosity ν in both cases and since the two turbines must be operating under similar hydraulic conditions, we have

$$\frac{R'}{R} = \frac{D'\sqrt{H'}}{D\sqrt{H}}$$

Thus the ratio of losses $1 - \eta$ and $1 - \eta'$ must be a function of the diameters and of the heads.

Ackeret offers the following solution† which is based, not on overall efficiencies, but on the hydraulic efficiencies of the turbine η_h and η_h' respectively:

$$\frac{1-\eta_h}{1-\eta_h'} = \tfrac{1}{2}\left[1 + \left(\frac{D'}{D}\right)^{1/5}\left(\frac{H'}{H}\right)^{1/10}\right]$$

The hydraulic losses $1 - \eta_h$ and $1 - \eta_h'$ differ from the overall losses $1 - \eta$ and $1 - \eta'$ by excluding the bearing and stuffing-box frictions, windage, etc. The difference may be of importance because the similarity between the model turbine and the prototype usually does not extend to bearings, shafts, stuffing boxes, and generally purely mechanical devices.

The Moody formula assumes that these losses are sufficiently similar to make the distinction between *hydraulic* efficiency and *overall*

† See: E. MÜHLEMAN: "The Scale Effect in the Efficiency of Reaction Turbines." *Schweizerische Bauzeitung*, 12th June, 1948.

efficiency an unnecessary refinement for the practical determination of the scale effect.

The Ackeret formula is developed on the assumption that the hydraulic losses derive from two causes:

(a) Kinetic losses which remain unaffected by the scale,
(b) Friction losses which vary with the Reynolds number.

The scale effect from Moody or Ackeret is strictly applicable to the point of best efficiency, which is the most important one for practical purposes. The scale effect at other points has not been investigated. It is generally assumed that the step up in efficiency η' to η calculated for the point of best efficiency is applicable at all guide-vane openings and speeds.

In consequence of the increase in efficiency η' to η, the hydraulic conditions are as though the effective head on the turbine were altered in the ratio η/η' and, therefore, all velocities and discharges are subjected to an adjustment of $\sqrt{(\eta/\eta')}$ whilst the outputs are affected in the ratio $(\eta/\eta')^{3/2}$

For impulse turbines, no scale effect is observed. Probably this is due to the deterioration in smoothness of the jet when the head increases —deterioration which nullifies any benefit that could be expected from reduced friction losses.

9. Mechanical Similarity.

$D = D'$
$H \neq H'$

As shown in § 3, two identical turbines under heads H and H' operate under similar *hydraulic conditions* when their speeds n and n' are in the ratio

$$\frac{n}{n'} = \sqrt{\frac{H}{H'}}$$

The hydraulic pressure over any given part of the turbine produces a force proportional to the head.

If P is the force resulting from hydraulic pressure over a surface F under the head H, and P' the corresponding force caused by the head H' over the same surface F, then

$$\frac{P}{P'} = \frac{H}{H'}$$

The centrifugal forces are proportional to the square of the speed, namely $(n/n')^2$ which is equal to H/H'. Thus, the centrifugal forces are in the same ratio as the hydraulic forces and, since the two runners are of

identical size, all stresses are proportional to the heads under which the turbines operate. The same applies to the specific pressures in bearings.

Consider now two turbines geometrically similar, therefore of same specific speed, but of different diameters D and D^* operating *under the same head*. By definition, these machines are geometrically similar in all their parts. For similar hydraulic conditions their speeds are in inverse ratio of the diameters. Therefore $n^*/n = D/D^*$.

H=H'
D≠D'

The hydraulic forces P and P^* acting over the geometrically similar areas F and F^* are in the ratio

$$\frac{P^*}{P} = \frac{F^*}{F} = \left(\frac{D^*}{D}\right)^2$$

Centrifugal forces are the products (mass $\times$ radius $\times$ ω^2). Provided the material from which they are made is of the same specific weight, similar parts of the turbines of diameters D and D^* are subjected to centrifugal forces C and C^* in the ratio

$$\frac{C^*}{C} = \left(\frac{D^*}{D}\right)^3 \cdot \frac{D^*}{D} \cdot \left(\frac{n^*}{n}\right)^2 \text{ which reduces to } \frac{C^*}{C} = \left(\frac{D^*}{D}\right)^2$$

The centrifugal forces are therefore in the same proportion as the hydraulic forces $C^*/C = P^*/P = F^*/F$.

The stresses are the quotient of the sum of these forces divided by the areas over which they apply, namely

$$\text{stress} = \frac{P}{F} + \frac{C}{F} \quad \text{and} \quad (\text{stress})^* = \frac{P^*}{F^*} + \frac{C^*}{F^*}$$

Since

$$\frac{P^*}{F^*} = \frac{P}{F} \quad \text{and} \quad \frac{C^*}{F^*} = \frac{C}{F}$$

it follows that the stresses are the same in both cases. Hence turbines of geometrically similar design are equally stressed when operating under the same head, irrespective of their size, so long as they operate under similar hydraulic conditions.

It follows that the stresses in turbines geometrically similar and made of the same materials are proportional to the head under which they operate, irrespective of their size, when they operate under similar hydraulic conditions.

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CHAPTER III

Francis Turbines

REVISED BY R. V. MATHEWS, ORIGINAL AUTHOR P. E. DÉRIAZ

1. General Characteristics.

By " Francis " turbine is understood a water turbine where the runner receives the water under pressure in a radial inwards direction and discharges it in a substantially axial direction (fig. 3.1).

Owing to the rotation of the runner a centrifugal force is imparted to the water which opposes the inward flow. The pressure at the runner inlet must be capable of overcoming this centrifugal force. Conditions

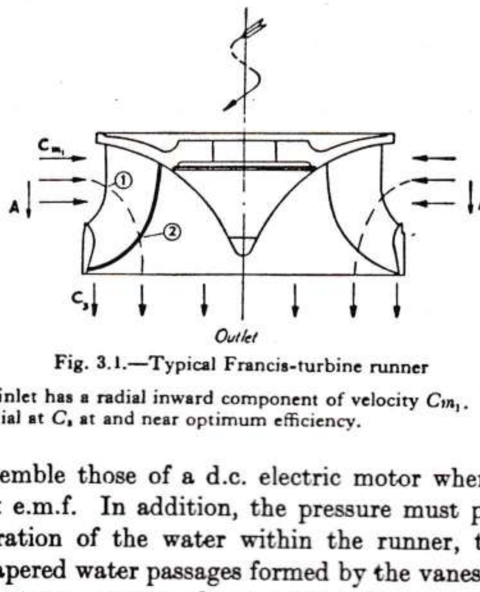

Fig. 3.1.—Typical Francis-turbine runner

The water at inlet has a radial inward component of velocity Cm_1. The outlet velocity is substantially axial at C_3 at and near optimum efficiency.

somewhat resemble those of a d.c. electric motor where the rotation causes a back e.m.f. In addition, the pressure must produce the required acceleration of the water within the runner, to discharge it through the tapered water passages formed by the vanes (fig. 3.2). The fact that the water pressure drops within the runner identifies the Francis turbine as a reaction turbine. The degree of reaction is defined as the ratio of the pressure difference between points 1 and 2 to the net head under which the turbine operates

Any turbine where the water velocity accelerates relatively to the runner is a reaction turbine. This condition can occur with an axial-flow (Kaplan or propeller) runner as well as with a Francis runner. All reaction turbines are not, therefore, Francis turbines, but all Francis turbines are of reaction type.

The name " Francis " is accepted to-day to cover all variations of turbines which conform to the broad definition above. " Francis " is the name of the designer who first built and tested a turbine of this character having radial inward entrance flow. He called it the " centre-vent water wheel ", meaning that the runner discharges centrally.

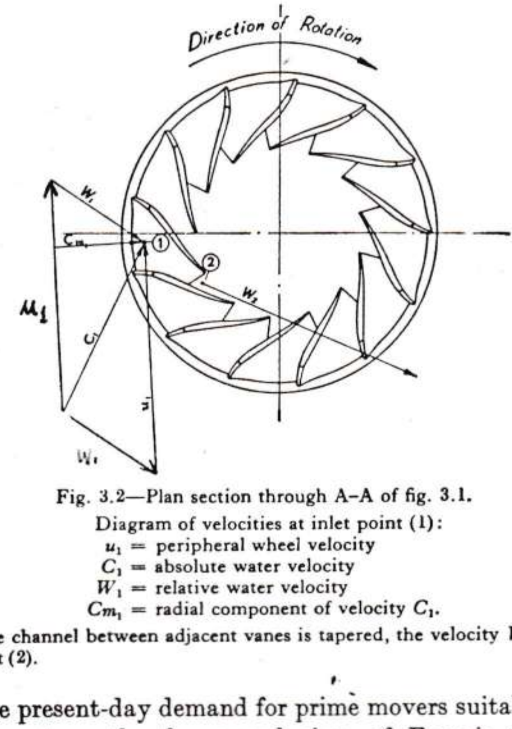

Fig. 3.2—Plan section through A–A of fig. 3.1.
Diagram of velocities at inlet point (1):
u_1 = peripheral wheel velocity
C_1 = absolute water velocity
W_1 = relative water velocity
Cm_1 = radial component of velocity C_1.

Because the channel between adjacent vanes is tapered, the velocity W_1 is increased to W_2 at exit point (2).

With the present-day demand for prime movers suitable for driving electric generators, the forms and sizes of Francis turbines have developed in a way which could not have been foreseen during Francis's lifetime. The demand has been for ever-increasing power per unit with resulting increase in size, leading at first to reduction in rotational speed. More economical machines were designed by increasing the specific speed (see definition Chap. I, § 4), which led to the present-day form of Francis turbine with its advantages and limitations, as described hereafter.

As Francis turbines are of the reaction type the water must enter the runner under pressure. This calls for sealing arrangements to avoid escape of water at the joint between rotating and stationary parts. Leakage causes an appreciable loss, even with low heads, and becomes important at high heads. This loss does not occur in an impulse turbine. On the other hand, the water velocity relative to the runner in a reaction turbine is small compared to that in an impulse runner. Thus there is a considerable reduction in friction losses, so that a well-designed reaction turbine has a best efficiency generally superior to that of an impulse turbine.

One of the alleged advantages of an impulse turbine is that it is less subject to wear in the case of silt-laden water. This claim has not been based on strictly comparable conditions, and the opinions of specialists on this point are considerably at variance. The lower relative velocity within the reaction runner tends to lessen wear. Again, the nature of the wear is such as to produce more irregular shapes of worn surface in the impulse turbine, where the water is more free to diverge from its desired course because it is not contained in channels under pressure. This leads to a greater deterioration of efficiency in an impulse wheel than in a reaction wheel, for the same weight of metal removed by erosion.

Much also depends on the nature of the eroding silt, the frequency of occurrence of high silt content in the water, and the load factor of the plant.

Concerning the eroding effect of sand or silt on the turbine, it must be stated that a well-designed hydro-electric installation should take into account not only the stoppages demanded by renewals of worn parts in the turbine, but also, and even more important, the gradual deterioration in storage capacity arising from the silting up of reservoirs. This presents for the civil engineer serious problems of sand elimination. Loss of reservoir capacity is more costly than the replacement of turbine parts and is much more difficult to rectify. Since upon elimination of sand depends the useful life of the hydro-electric scheme, the resistance to sand erosion by the turbine must be considered to be of lesser importance than the achievement of the highest efficiency possible. For this reason, the Francis turbine should be given preference where conditions permit an equally economical installation of either type of turbine. Conditions which allow this choice usually occur with large heads.

For equal power and head, the Francis turbine requires a smaller space for its installation. It also has the advantage that highest efficiency is obtained near full load. This compares favourably with the

impulse turbine, which works at its best efficiency at or near half the rated load, with the efficiency falling off gradually towards full load. Thus the Francis turbine is more efficient where most energy is generated, but it is less efficient at small loads.

In the past, it has been the custom to sub-divide Francis turbines into classes according to the manner in which the water was admitted. Thus, open-flume turbines were treated as distinct from spiral-cased turbines. Again, according to the number of wheels on the same shaft the turbine was single or twin, with outlets to a common draft tube or inlets from a common spiral casing. Descriptions of these variations can be found in earlier textbooks. They are of interest in affording a knowledge of the very wide variety of machines which have been built and operated in the past.

It is not intended to describe obsolete arrangements here, and consideration will be given only to the most modern types which have superseded all others owing to (*a*) their technical and economic advantages; (*b*) their suitability to concentrate large outputs on one shaft, operating at speeds best suited to powerful electric generators. Therefore only the Francis turbines in spiral casings, as shown generally in figs. 3.3, 3.5, 3.6 and 3.8 will be considered.

2. Turbine Arrangements.

Francis turbines can be arranged in two ways: with vertical shaft, as shown in fig. 3.3, or with horizontal shaft as shown in fig. 3.5.

The vertical-shaft arrangement requires the minimum space for installation and therefore permits the smallest area of power house. It is not only more economical in space, but in many cases it is the only practical solution for large machines, especially when the topographical nature of the site limits the size of the power house.

In schemes where the building of a dam across a narrow gorge requires provision of a wide spillway for dealing with floods, it will often be found that the area available for the power house is very restricted. The vertical-shaft arrangement then becomes imperative.

Hydraulically, the vertical-shaft arrangement is preferable to the horizontal, as it permits the placing of the turbine at a lower level relative to the tailwater level with a minimum of deep excavation. This is important in order to reduce the liability to erosion of the runner by the cavitation which results from too low an absolute pressure on the discharge side of the wheel. From the consideration of manufacture, horizontal-shaft turbines are only economical for relatively small-size machines. With large sizes of spiral casing, complicated

1. Return-motion linkage.
2. Guide-vane servomotor
3. Top cover
4. Stuffing box
5. Main shaft
6. Turbine guide bearing housing
7. Regulating ring
8. Breaking links
9. Spiral casing
10. Guide vane
11. Sealing rings
12. Runner
13. Air inlet valve
14. Throat ring
15. Draft tube liner
16. Removable draft tube cone for underneath dismantling of runner
17. Speed ring
18. Stay vane
19. Pivot ring
20. Thermocouple for bearing-pad temperature
21. Guide bearing pad
22. Guide-bearing oil sump
23. Main distributor valve for governor
24. Air-inlet pipe for depressing water in draft tube for synchronous-condenser operation
25. Cooling-water inlet pipe for seals during synchronous condenser operation
26. Draft tube manhole
27. Guide-vane lever
28. Connection to pilot governor

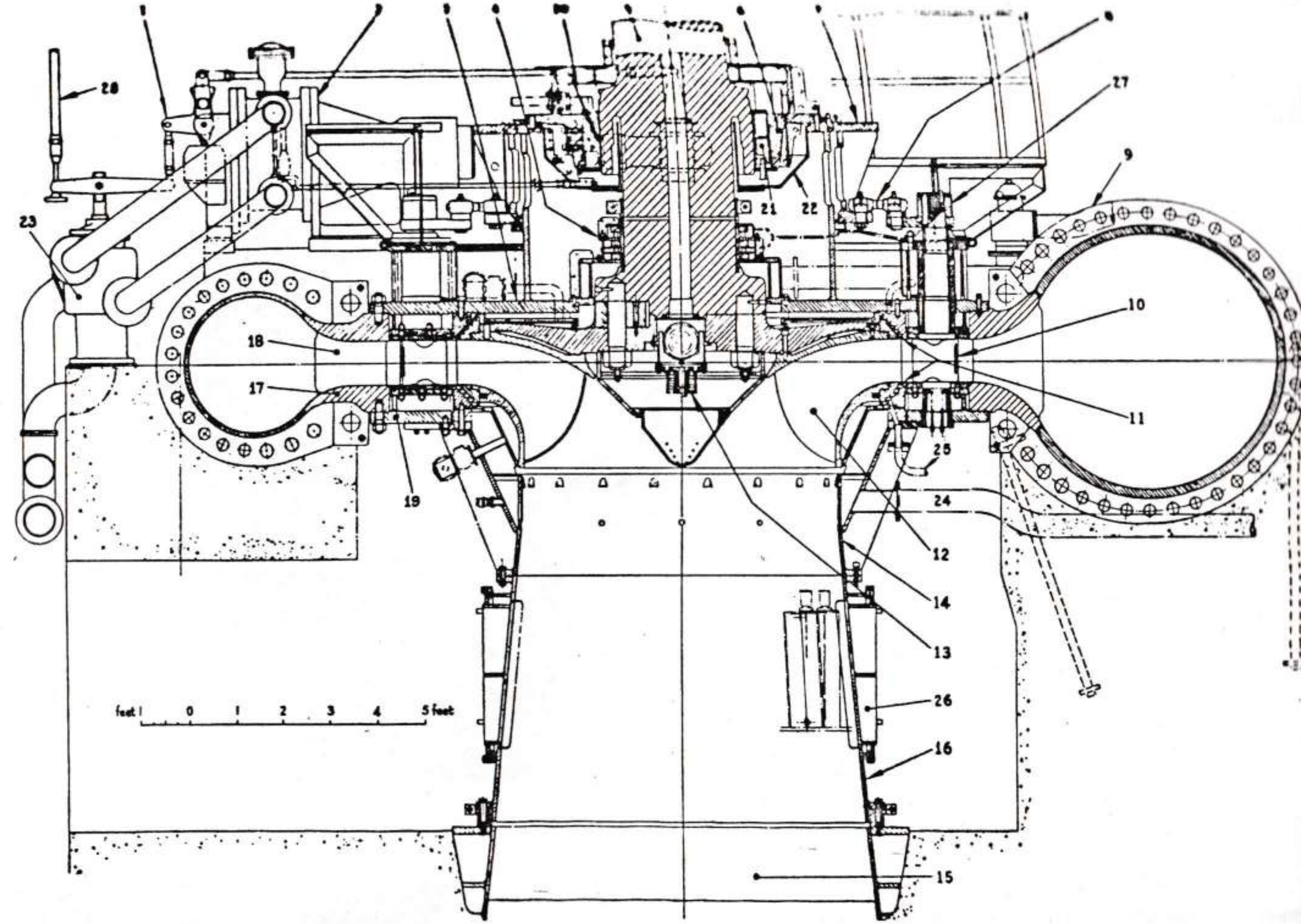

Fig. 3.3.—Vertical-shaft Francis turbine for Tumut 1 power station of Snowy Mountains Hydro-Electric Authority. Designed for 110,500 h.p. under 1,000 ft., speed 375 r.p.m.

English Electric

structures would be necessary to support the spiral casing and the draft tube, unless they were made very heavy.

For the vertical arrangement the larger parts of the turbine are embedded in concrete, which not only affords unyielding support but also prevents distortion and increases the resistance to vibration.

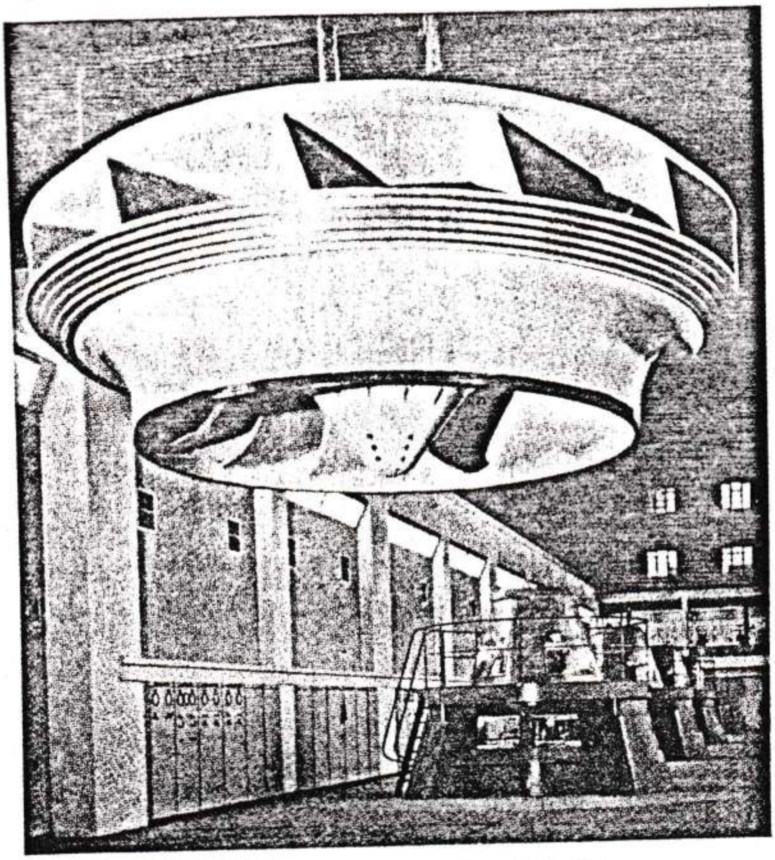

Fig. 3.4.—Runner for turbine shown in fig. 3.3.
Typical high head Francis runner of low specific speed; fabricated construction; stainless steel.

English Electric

The horizontal turbine has the advantage of providing greater accessibility to all parts of the set. Provided the physical size permits such arrangement, the runner can be dismantled after removal of part of the draft tube with more ease than with a vertical-shaft machine. The importance of such considerations depends largely on the extent of

the repairs or maintenance to the turbine runner which may be expected. When several machines are operating in the same power house, routine maintenance requires that an overhaul should take place at least every five years. This applies to the generator as well as to the turbine. Since the removal of the generator rotor affords the most convenient access to the turbine, it is evident that for normal maintenance work the vertical-shaft arrangement is nearly as convenient as the horizontal. This assumes that, with water reasonably free from silt and a well-designed turbine, maintenance should not require more frequent dismantling of the turbine than of the generator. Even with a vertical shaft, provision can be made for removal of the runner without dismantling more than the upper section of the draft tube (fig. 3.3).

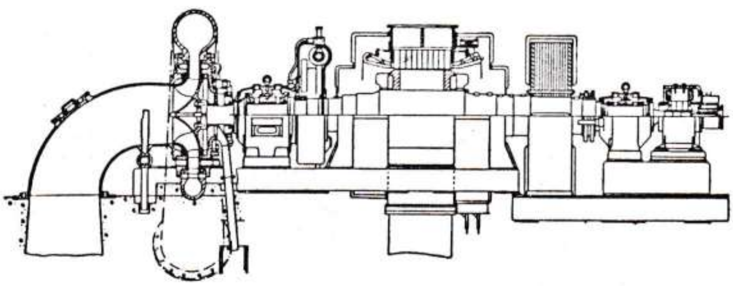

Fig. 3.5.—Horizontal-shaft Francis turbine with generator, flywheel and exciter as installed at Victoria Falls (Northern Rhodesia). 1450 h.p. under 346 ft. head at 1000 r.p.m.
The great distance from the Falls to any important centre of industry explains the small demand for power which was the deciding factor for selection of the horizontal-shaft setting best suited to small-size machines.

It is obvious that the daily work of lubrication on all guide-vane bearings is greatly facilitated by the vertical-shaft setting. The modern thrust bearing can conveniently carry all loads arising from rotor weight and hydraulic thrusts. It is best arranged submerged in an oil bath, which is only possible with the vertical-shaft arrangement. The thrust bearing cannot be dispensed with in the horizontal machine; because there is reversal of hydraulic thrust under certain conditions of load, it is necessary to provide a double thrust bearing.

In the vertical-shaft arrangement, it is advisable to jack up the rotor and lift the thrust collar off the thrust-bearing pads before starting. The journal bearings, however, become mere guides without any theoretical loads, since all forces balance by reason of symmetry. With a horizontal shaft heavy rotors require jacking up at each journal bearing. A further disadvantage of the horizontal arrangement is the

necessity for an additional bend on the discharge side, to lead the water from the horizontal to the vertical direction, and then again into the horizontal direction of the tailrace. Thus the efficiency of horizontal turbines is recognized to be generally inferior to that of vertical turbines by one or two per cent. With the present tendency towards an increase in the size of units and with the simultaneous demand for high efficiencies and reduction of area of the power house, the vertical-shaft

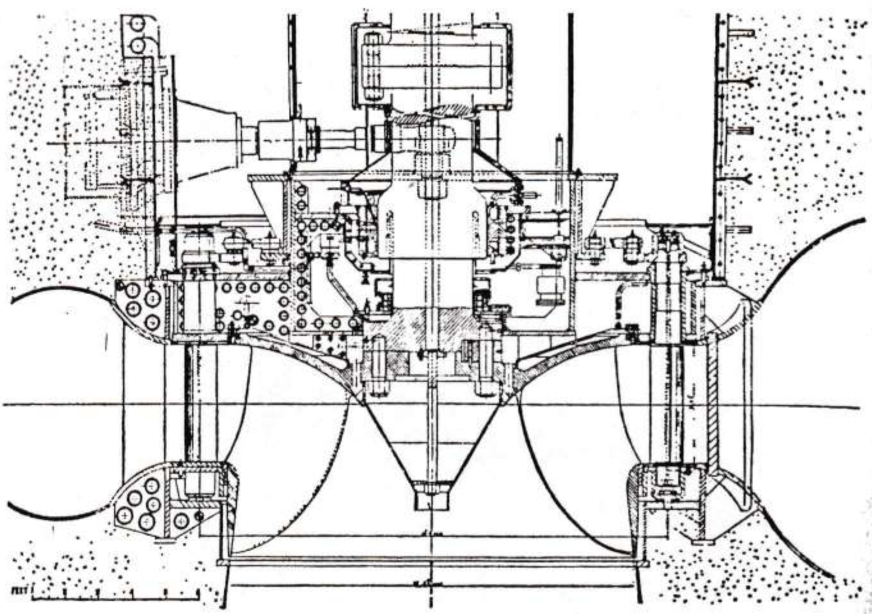

Fig. 3.6.—Vertical-shaft turbine at Rihand, Uttar Praedesh, India. 77,000 h.p. under 225 ft. at 150 r.p.m.

English Electric

arrangement is usually the more favourable. When small machines are to be installed for high heads, the horizontal-shaft arrangement may be preferred, but even then the trend is in favour of vertical-shaft machines. In certain parts of the world where designers of turbines are dealing with many, usually very large, units the horizontal-shaft arrangement is regarded as obsolete. In favour of the horizontal-shaft arrangement, however, it can be noted that the foundations are simpler, but this is valid only for comparatively small units.

3. Turbine Runner.

3.1. *Factors Affecting Design.*

The most important item in the turbine is the runner (figs. 3.4, 3.7 and 3.9) which must be designed to give the best performance under all conditions prevailing at the particular site. The whole turbine must be considered to be dependent upon the runner with regard to size and position. All parts of the turbine are built round the runner and are designed to lead the water to it and away from it in the manner most suitable for obtaining maximum efficiency.

Fig. 3.7.—Runner casting for turbine illustrated in fig. 3.6. Labyrinth seal rings have not yet been shrunk on. The stainless steel protection on the blades can be clearly seen.

English Electric

The best efficiency is reached under conditions where the total losses in the whole turbine are a minimum. Losses arise from the following causes:

(*a*) Velocity head of the water at discharge from the runner is only partly recovered in the draft tube.

(*b*) Skin friction in the runner passages.

(*c*) Eddies at runner-blade inlet.

(*d*) Eddies at runner-blade outlet.

(*e*) Water leakage past the seals between moving and fixed parts of the turbine.

(*f*) Friction losses in the bearings and shaft gland.

(*g*) Friction losses in the spiral casing, guide apparatus, and draft tube.

Of the above losses, the two first-mentioned (*a*) and (*b*) are directly dependent on the size of the runner. Those under (*c*) and (*d*) can be very small with correct inlet angles and blade shapes. All the others arise from causes outside the runner.

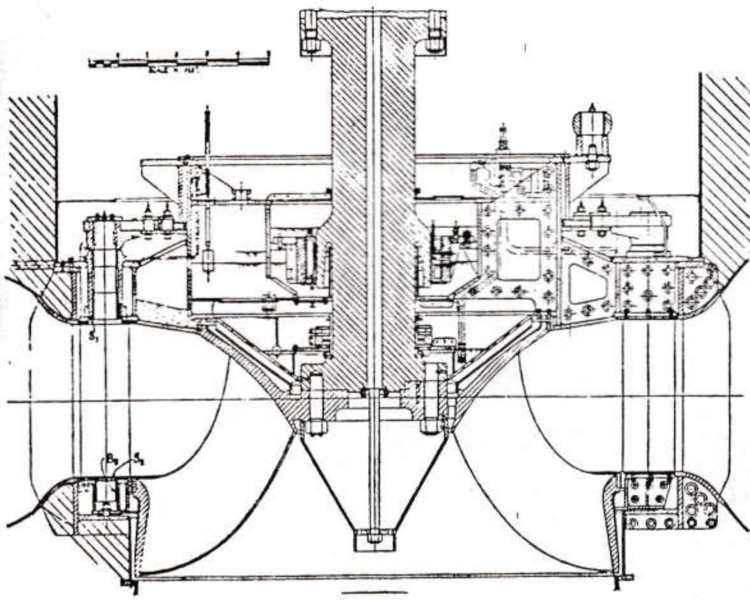

Fig. 3.8.—62,000 h.p. vertical Francis turbine for Ponte Coberta power station, Brazil.

115.4 r.p.m. under 123 ft. head.

English Electric

The runner must be designed so as to obtain a minimum sum of losses (*a*) and (*b*) and this is the basis upon which the size of the runner is fixed.

The losses (*a*) are reduced by adopting small absolute velocities at the discharge, for which a large exit diameter D_2 is necessary. An increase in D_2, however, entails a correspondingly large area of the blades and smaller exit angles. Not only does this result in a larger surface for skin friction, but also the velocity of the water relative to the blade increases with the peripheral speed of the runner. The skin friction increases as the square of the relative water velocity over the

blade area, which itself increases as the square of the exit diameter. The losses (*b*) therefore increase very rapidly with increase of D_2. Conversely, to reduce the skin-friction losses smaller runner diameters are required and, therefore, larger exit velocities with larger losses. These are proportional to the square of velocity which itself increases in inverse proportion to the square of the diameter.

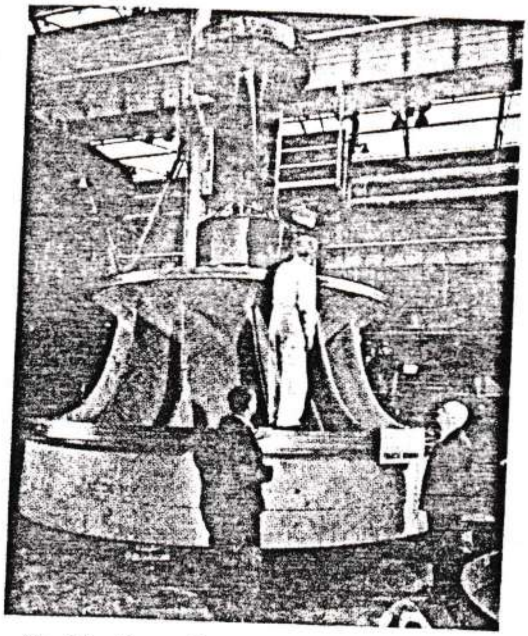

Fig. 3.9.—Runner for turbine illustrated in fig. 3.8. The sealing rings have already been shrunk into place.

English Electric

The best compromise is reached when suitable exit angles are chosen, and practice indicates that this is so when the value of $D_2^3 n/Q_\perp$ lies between 85 and 115, depending on the finish of the runner blades and the efficiency of the draft tube. The symbol $\perp$ means that the direction of the discharging water is perpendicular to the direction of rotation of the runner. This ratio is applicable to all reaction turbines and can be derived by considering a quantity of water $Q_\perp$ passing with uniform axial motion through a cylinder of diameter D_2 rotating at n r.p.m. The tangent of the angle β_2 of the water path relative to the cylinder is proportional to $D_2^3 n/Q_\perp$ (see fig. 3.10). ([illegible])

The quantity $D_2^3 n/Q_\perp$ is dimensionless. For practical use the speed is expressed in revolutions per minute to which the values 85 to 115

correspond. The diameter D_2 and the discharge $Q_\perp$ are then in feet and cubic feet per second respectively. It will be noted that the expression is independent of the head under which the turbine operates and consequently is also independent of the specific speed. For the designer this consideration is of appreciable help as it permits the laying out of the runner blading in a systematic way which will be outlined hereafter.

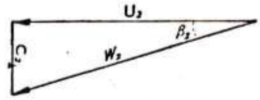

Fig. 3.10.—Velocity diagram at exit and at periphery of runner (Flow Line I of fig. 3.12)

$$C_{2\perp} = \frac{Q_\perp}{\frac{1}{4}\pi D_2^2} \qquad U_2 = \pi D_2 \frac{n}{60}$$

$$\tan \beta_2 = \frac{C_{2\perp}}{U_2} = \frac{Q_\perp}{\frac{\pi}{4} \times \frac{\pi n}{60} \times D_2^3}$$

Therefore $$\frac{D_2^3 n}{Q_\perp} = \frac{240}{\pi^2 \tan \beta_2}$$

3.2. *Determination of Runner Dimensions.*

In Chap. I it is explained how to make the choice between Pelton, Francis, fixed-blade propeller, and Kaplan turbines. The number of units to be installed to deal with the available water has also been considered, together with the speed at which the machines are to run.

The following data are therefore available, for which the turbine must be designed:

The net head H_e —given by site conditions.
The discharge $Q_{1/1}$—at full load, as derived from the output and efficiency.
The speed n —according to the permissible specific speed and suitability to the generator.

We must also know at what fraction of the full load the best efficiency is required. The corresponding discharge can be calculated and will be called Q_{optimum}.

The discharge for which the runner blading is laid out is $Q_\perp$ at which the water leaves the runner without rotational component. This discharge is slightly larger than that which gives the highest efficiency and, from Q_{optimum}, we can derive $Q_\perp$ by writing $Q_\perp \simeq 1{\cdot}04 \times Q_{\text{optimum}}$. The runner exit diameter D_2 is now obtained from

$$\frac{D_2^3 n}{Q_\perp} = 85 \ldots 115.$$

Knowing the specific speed $n_s = \frac{n}{H}\sqrt{\left(\frac{N}{\sqrt{H}}\right)}$ fig. 3.11 gives suitable coefficients for determining the principal proportions of the runner, such as the inlet diameters D_1 external and D_1 internal and width of the guide-vane apparatus B_0. These coefficients have been derived from turbines actually built and can be found in the technical literature on the subject.

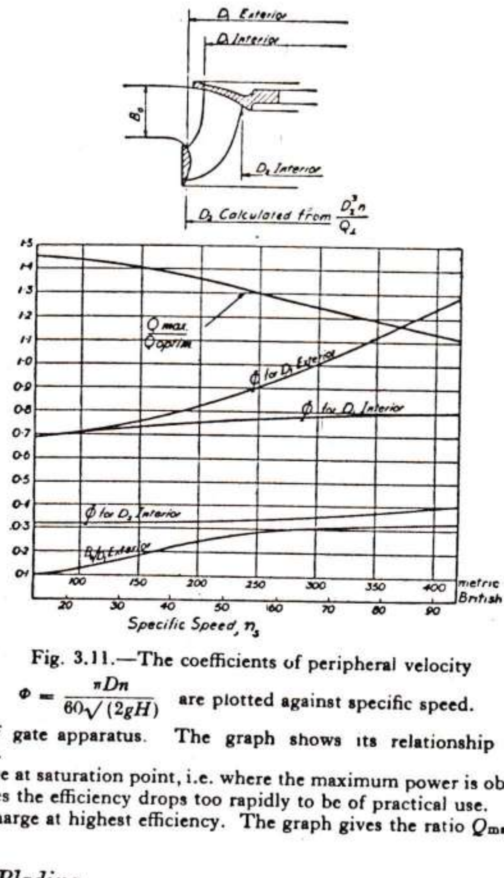

Fig. 3.11.—The coefficients of peripheral velocity

$$\Phi = \frac{\pi D n}{60\sqrt{(2gH)}}$$ are plotted against specific speed.

B_0 = width of gate apparatus. The graph shows its relationship to the diameter $D_{1\,\text{exterior}}$.

Q_{max} = discharge at saturation point, i.e. where the maximum power is obtained. At larger discharges the efficiency drops too rapidly to be of practical use.

$Q_{optimum}$ = discharge at highest efficiency. The graph gives the ratio $Q_{max}/Q_{optimum}$.

3.3. *Runner Blading.*

Having laid out the approximate outline of the runner the flow lines are drawn as indicated by the Roman figures I–V in fig. 3.12. Velocity diagrams at exit are now drawn for each of these flow lines. For the flow line I which is next to the rim of the runner, the velocity diagram is similar to fig. 3.10, but the velocity C_2 must be taken inside the blading, therefore the space occupied by the blades must be deducted

to give the true area of discharge. For the other flow lines the velocity C_2 is the same, assuming a regular and uniform discharge across the whole exit edge of the runner blade.

Starting from the exit edge, sections are drawn as indicated on the right-hand side of fig. 3.12, where the section of the runner blade through a cone CC is shown in development, for the middle flow line III. These blade sections are laid out to conform with the velocity angles of the diagrams.

The next blade is now drawn in the development CC and the pitch chosen to give adequate proportions for the channel formed between

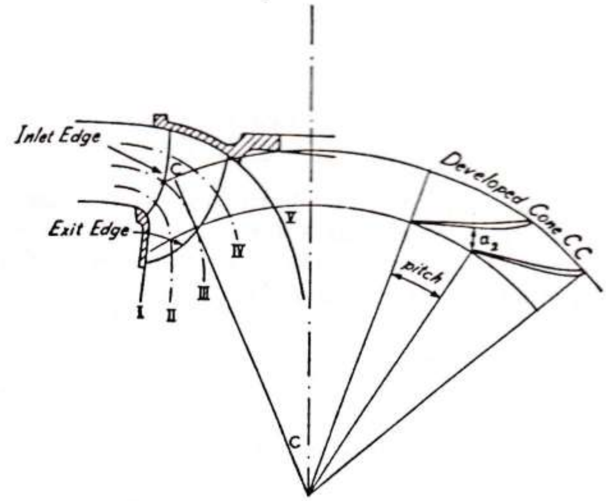

Fig. 3.12.—Flow lines I to V are laid down in the runner profile and sections of vanes drawn on the developed cone CC for the flow line III

the two adjacent blades. The pitch is related to the number of blades which usually is such that there is no common factor with the number of guide vanes in order to avoid setting up resonant vibrations. As the number of guide vanes is chosen to be even for construction reasons, the number of runner blades is as a rule odd.

Before going further, a check is made for the required discharge Q_1 which must be obtained by integrating the products of exit velocity W_2 and the width of passage a_2 along the length of the exit edge from flow line I to flow line V and then multiplying by the number of blades.

Having thus established, as a first approximation, the form of the blades at exit, the surface of the blades is extended upstream in a gradual curve, paying attention to the necessary tapering of the channel between adjacent blades to produce the nozzle effect by which the water is

gradually and sufficiently accelerated through the runner with the minimum of eddies.

The blade is thus extended toward the inlet and the edge is so located that the water flows smoothly on to the blade surface as laid out. The relative-velocity angle at the blade inlet is determined from the inlet diagram based on the Euler equation.

The Euler equation is obtained by equating the change in the moment of momentum of the water as it passes through the runner to the torque delivered to the turbine shaft.

Let C_{u_1} = the water velocity component tangential to the runner rotation,
r_1 = the radius, or distance from the turbine centre line to the point where C_{u_1} is measured.
g = the gravitation constant.
H = the net head.
ϵ = the hydraulic efficiency of the runner.
ω = the angular velocity of the runner $= \pi n/30$.

The volume of water Q of specific gravity γ passing in each second has a velocity component C_{u_1} in the direction of rotation (whirl component) changed into the velocity component C_{u_2}. In each second the mass $\gamma Q/g$ is therefore subjected to a change in moment of momentum

$$\frac{\gamma Q}{g}(C_{u_1}r_1 - C_{u_2}r_2) = \text{shaft torque in lb. ft.}$$

On the other hand the runner output is

$$N = \gamma Q H \epsilon \quad \text{lb. ft. per sec.}$$

from which the shaft torque can be derived as N/ω.

Then
$$\frac{\gamma Q}{g}(C_{u_1}r_1 - C_{u_2}r_2) = \frac{\gamma Q H \epsilon}{\omega}$$

and
$$C_{u_1}r_1 - C_{u_2}r_2 = \frac{gH\epsilon}{\omega} \quad \text{(Euler equation).}$$

For the purely axial discharge $Q_\perp$ the residual momentum in the direction of rotation of the water when leaving the runner is nil, ($C_{u_2} = 0$), therefore the moment of the momentum at runner inlet must equal the shaft torque. For this particular case, the equation of Euler takes the simple form

$$C_{u_1}r_1 = \frac{gH\epsilon}{\omega}.$$

The hydraulic efficiency ϵ differs from the turbine efficiency η to the extent that it does not include the losses (*e*), (*f*) and (*g*) in § 3.1. ϵ can usually be taken as $\eta + 2\%$ as a first and sufficient approximation. The expression in the right half of the equation $gH\epsilon/\omega$ is thus given completely by design data.

The water approaches the runner in a free vortex where $C_{u_1}r_1$ is a measure of its intensity. For a frictionless flow $C_{u_1}r_1$ is constant whatever the radius r_1.

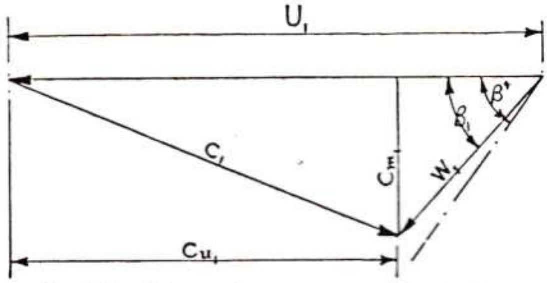

Fig. 3.13.—Velocity diagram at runner inlet for Q_1

$U = r_1\omega$ $\qquad C_{u_1} = \dfrac{gH}{\omega r_1}$

$Cm_1 = \dfrac{Q_1}{2\pi r_1 B_1}$

C_1 = absolute velocity of approach before entering the blading.
W_1 = relative velocity of approach.
β_1 = mean flow angle relative to runner.
β^* = blade inlet angle.

At the inlet to the runner blade, the width of the water passage is B_1 and thus the axial component of velocity is

$$C_{m_1} = \frac{Q_1}{2\pi B_1 r_1}.$$

The peripheral velocity of the runner at the radius r_1 is

$$U_1 = r_1\omega$$

and the diagram of inlet velocities can be drawn (see fig. 3.13).

From the inlet diagram (fig. 3.13) can be read the relative inlet velocity W_1 in magnitude and in direction. Because the number of blades is finite, the blade inlet angle β^* is designed a few degrees steeper than β_1.

Until now, the dimension r_1 has been taken arbitrarily from fig. 3.11. The angle β^* may fall in line with the shape of blade surface laid out when starting from the exit edge, but an adjustment of the radius r_1 is

generally necessary to obtain compatibility. The adjustment is readily made, when considering that a small reduction in the radius r_1 causes simultaneously a reduction in U_1 and an increase in C_{u_1}. C_{m_1} is less affected by the change of r_1 and the resulting change to β_1. Thus, alteration in r_1 has a very rapid effect on the angle β_1 and the inlet edge of the blade can therefore be chosen so as to give the required smooth entry into the runner.

A similar procedure is applied to each flow line, and the complete surface of the blade is obtained by ensuring continuity of form when passing from one flow line to another.

At this stage attention must be given to the form of the blade as regards the phenomenon of cavitation.

3.4. *Cavitation.*

Cavitation develops where and when the absolute pressure drops to or below the vapour pressure of the water. Fig. 3.14 shows a typical development of blade section along a flow line such as I of fig. 3.12 and

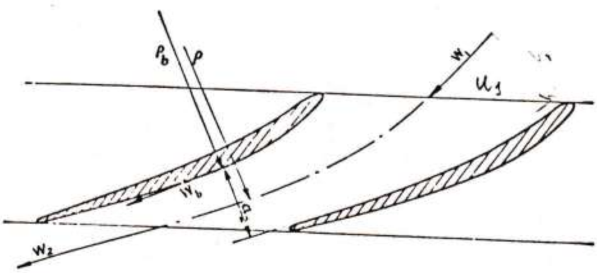

Fig. 3.14.—Check for safety from cavitation

W_b = water velocity relative to runner at back of blade.
ρ_b = radius of curvature of back of blade along flow line I.
W_2' = mean exit relative velocity as given by fig. 3.10.
a_2 = width of water passage at runner exit on flow line I.

Assuming a free vortex flow

$$W_b \rho_b = W_2(\rho_b + \tfrac{1}{2}a_2)$$

i.e.

$$W_b = W_2\left(1 + \frac{a_2}{2\rho_b}\right)$$

gives the relation between curvature, width of channel, and relative velocity, by comparing the flow issuing at a_2 to a free vortex flow relative to the runner, with its centre at the centre of curvature of the back of the blade. The pressure in this zone exceeds the vapour pressure of water by

$$H_B - H_s - \eta_D \frac{C_2^2}{2g} - \frac{W_b^2 - W_2^2}{2g}$$

where H_B is the atmospheric pressure minus the vapour pressure of water in feet;

H_s is the elevation of the runner above tailwater level, in feet;

η_D is the efficiency of the draft tube;

C_2 is the absolute water velocity at runner exit.

This expression must always *remain positive if cavitation is to be avoided,*

therefore
$$\frac{H_B - H_s}{H} > \eta_D \frac{C_2^2}{2gH} + \frac{W_b^2 - W_2^2}{2gH}.$$

The left-hand term of this expression is the "sigma" introduced by Thoma, which must remain larger than the *critical* value at which cavitation starts:

$$\sigma_c = \eta_D \frac{C_2^2}{2gH} + \frac{W_b^2 - W_2^2}{2gH}.$$

By using the relation of fig. 3.14

$$W_b^2 = W_2^2 \left(1 + \frac{a_2}{\rho_b} + \frac{a_2^2}{4\rho_b^2}\right)$$

in which the last term can be omitted since ρ_b is large when compared with a_2.

$$W_b^2 - W_2^2 = \frac{a_2}{\rho_r} W_2^2$$

and replacing the velocities by their ratio to $\sqrt{(2gH)}$ as

$$c_2 = \frac{C_2}{\sqrt{(2gH)}} \quad \text{and} \quad w_2 = \frac{W_2}{\sqrt{(2gH)}}$$

the useful relation is obtained:

$$\sigma_c = \eta_D c_2^2 + \frac{a_2}{\rho_h} w_2^3.$$

The term $\eta_D c_2^2$ depends on the exit diameter of the runner which has already been chosen, and on the efficiency of the draft tube. w_2 is likewise determined by the velocity diagram. The values of a_2 and ρ_b must thus conform with the conditions of $(H_B - H_s)/H \geqslant \sigma_c$ in order to prevent cavitation.

Larger values of a_2/ρ_b mean greater risk of cavitation and a limiting value of a_2/ρ_h is reached beyond which cavitation definitely occurs. This condition fixes the proportions of the water passages, and the blade curvature. A similar method of calculation, but somewhat more involved, is used to check the margin against cavitation for the curvature of the blade near the inlet edge.

The runner blading must now be examined for all conditions of operation and any adjustments made which are found to be necessary. The operating conditions include variable heads and variable loads. Often the final blade form must be a compromise between the ideals for conditions which are partly conflicting.

In fig. 3.15 a diagram of velocities shows the conditions for various loads at constant speed and head.†

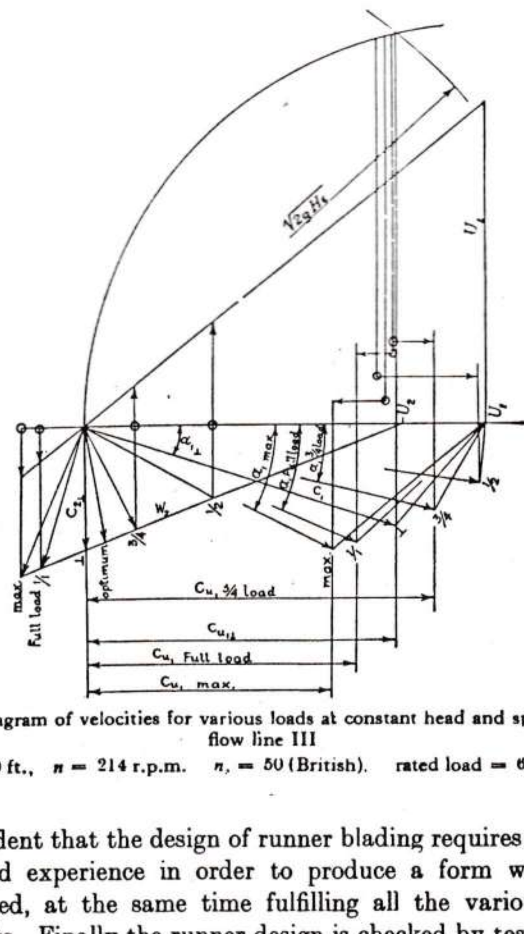

Fig. 3.15.—Diagram of velocities for various loads at constant head and speed for middle flow line III

H = 260 ft., n = 214 r.p.m. n_s = 50 (British). rated load = 62,000 h.p.

It is evident that the design of runner blading requires considerable practice and experience in order to produce a form which can be manufactured, at the same time fulfilling all the various hydraulic requirements. Finally the runner design is checked by testing a model turbine in a special hydraulic laboratory. These model tests comprise

† For more information about the graphical determination of such diagrams. reference should be made to the specialized works such as: CAMERER, *Vorlesungen über Wasserkraftmaschinen*, W. Englemann, Leipzig, 1924.

the determination of complete characteristic curves from which output and efficiency can be read for a whole range of heads and guide-vane openings, as well as the determination of the value of σ_c and of runaway speeds.

3.5. *Runaway Speed.*

It is customary in hydro-electric engineering to design all rotating parts to stand the full runaway speed that may occur should the governor for any reason fail to operate. Information about this runaway speed is obtained from the model tests.

Figs. 1.7–1.10 of Chap. I give the characteristics obtained with the model test turbine. Fig. 1.8 refers in particular to a Francis turbine. The discharge Q_1 is plotted against n_1, the speed under unit head for each guide-vane opening a_0. It will be noted that, for a medium-specific-speed Francis turbine to which this figure refers, the maximum runaway speed occurs at the maximum guide-vane opening. Furthermore, the discharge Q_1 falls off as the speed increases. (The discharge at the normal running speed is indicated by the dotted line.) This drop in the discharge at high speed is a result of the centrifugal action of the runner of the mixed-flow type as explained in § 1. The larger the difference in radius between inlet and outlet edges, the more pronounced will be this drop in discharge. Therefore, it will be found that for low-specific-speed runners the ratio of runaway speed to normal speed is considerably smaller than for higher specific speeds.

Fig. 1.1 of Chap. I shows the profiles of Francis runners for $n_s = 20$, 40 and 70. For the higher specific speeds, the flow through the runner becomes mostly axial, the centrifugal action is very restricted, and the discharge at high speeds is larger than the discharge at normal speed. The characteristic curves take a form shown in fig. 1.9; the Francis turbine shows the character of a fixed-blade propeller turbine, with the ratio of runaway speed to normal speed considerably higher than for a medium-specific-speed runner.

When the discharge is reduced at high speed as in fig. 1.8, some pressure surge may take place at the moment of runaway, and the resulting increased head must be taken into account. For a high-specific-speed turbine of character similar to that shown in fig. 1.9, the considerable increase in flow at runaway causes increased head losses in the pipeline and spiral casing. In all cases the generator windage losses have a braking effect to be taken into consideration. Runners working near the cavitation limit may show some increase in runaway speed because of the reduced fluid friction.

3.6. *Mechanical Strength.*

As outlined earlier the form of the runner blading is determined from the hydraulic data specified. Now the mechanical strength necessary to transmit the power from the vanes to the runner crown and to the turbine shaft and to resist hydraulic and centrifugal forces must be ascertained.

Analysis of stress distribution is lengthy and extremely complicated because of the peculiar shape of the vanes, the position of the forces involved, and the multiplicity of operating conditions. By application of the methods of mechanical similarity, data from previously made runners of known behaviour can be systematically applied to new designs. The mechanical similarity is outlined in Chap. I, § 9. It applies not only to runners but to any part of the turbine.

3.7. *Construction.*

Francis-turbine runners are very often one-piece castings. The material is generally cast steel, either mild steel or stainless steel, sometimes bronze, but seldom cast iron. Steel has the advantage that casting defects and normal wear can be made good by electric welding.

When conditions justify the cost, the runners are cast in stainless steel, usually of 12–14% chromium content.† This alloy shows considerably more resistance to erosion by cavitation or by sand than does ordinary cast steel.

To economize with the expensive alloy, the mild-steel casting can be protected locally by coating it with a stainless-steel deposit welded on electrically. Alternatively, it may be found to be more economical to weld on strips of stainless steel. The electrodes used for this work are usually 18% chromium and 8% nickel.

† Specification for stainless steel runner castings. B.S. 1630 grade A.
Chemical Composition.

Carbon	0·12% maximum
Silicon	1·00% ,,
Manganese	1·00% ,,
Nickel	1·00% ,,
Chromium	12–14% ,,
Sulphur	0·05% ,,
Phosphorus	0·05% ,,

Physical Properties (after complete heat treatment).

Ultimate tensile strength	Not less than 35 tons/sq. in.
Yield point	Not less than 22 tons/sq. in.
Elongation	Not less than 20% (B.S.I. Test Piece "C").
Bend test	Not less than 120° over 1 in. radius on a specimen 1 in. diameter or 1 in. wide × $\frac{3}{4}$ in. thick.

Sound castings are not always easy to obtain, so that instead of casting, the runners are often fabricated from pressed-steel blades welded to a cast or rolled rim and a cast hub. This permits pre-machining the rim and hub to smooth and accurate contours, and gives a blade surface that requires little hand-grinding to bring it to final finish. The blades are profiled by machining flat before pressing and can be protected in places by stainless steel if required. This is also conveniently done before pressing.

Since the runner is the part in which the maximum relative velocities occur, attention must be given to obtaining the required high degree of smoothness, as otherwise losses due to skin friction and eddies will cause a drop in efficiency. The best finish is required in the regions of high relative velocities, e.g. on the rim and at the exit side of the blades near the rim.

For low-head turbines of large dimensions, a high degree of finish is not essential in the neighbourhood of the hub nor on the pressure side of the blades adjoining the hub, because the relative velocities are low. In such places, it is sufficient to grind the high spots of the cast surface and to concentrate on producing a better finish where it is essential.

3.8. *Sealing Rings.*

The water that leaks through the clearance between the rotating runner and the fixed surrounding parts represents a loss which assumes important proportions when the operating head is large. To reduce these losses, the gap is often arranged in steps, as shown in fig. 3.8, for medium and low heads. For larger heads the seals can be formed as labyrinths, as in figs. 3.3 and 3.6, which combine restricted passages with many changes in direction to give additional resistance to the leaking water. The labyrinth seals are often formed by renewable forged-steel rings shrunk on the runner.

3.9. *Balance.*

As the last operation in manufacture, the runner is balanced by removing material from either the hub or the rim. Extra thickness is allowed in such places to cater for this procedure. Sometimes it is possible to correct the balance by addition of weights, but this is not always practicable or desirable. Static balance is always ensured and is sufficient in many cases, but dynamic balancing is very desirable. It will be appreciated that where large castings are involved, some parts can be widely different from the design thickness, and static balancing may give inadequate correction for the lack of symmetry.

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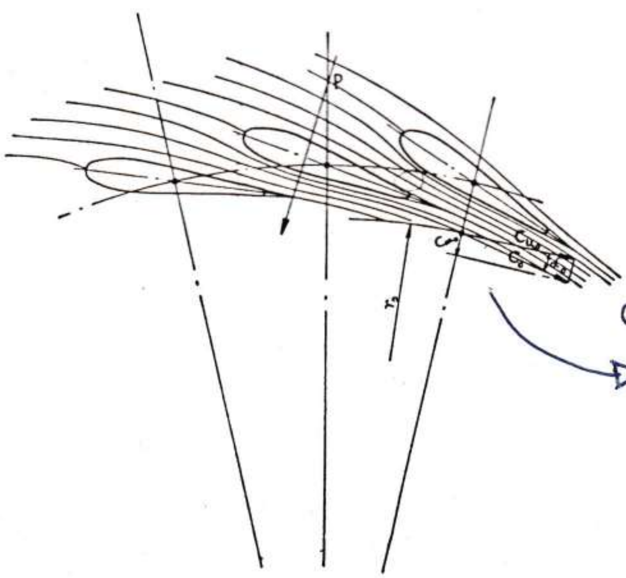
Dynamic balancing is carried out by rotating the runner when supported freely on a point slightly above its centre of gravity. Owing to its own inertia, it continues rotating and aligns itself on its dynamic axis. If this coincides with the geometrical axis of the runner, the balance is dynamically correct. If correction is necessary, it is effected by the removal of metal at two places to give the requisite correcting couple.

4. Guide-vane Apparatus.

4.1. *Design.*

Immediately in front of the runner are placed the guide vanes which permit the control of turbine output. A part section through the central plane of the guide vanes is shown in fig. 3.16, with flow lines.



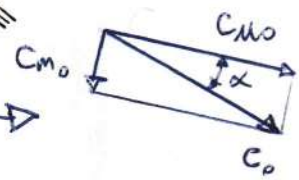


Fig. 3.16.—Section through guide-vane apparatus, half-open position

At the exit from the guide vanes the water has a velocity C_0 with the components C_{u_0} and C_{m_0}, which result at the inlet to the runner in the velocities C_{u_1} and C_{m_1} shown in fig. 3.13.

The angle α_0 of the absolute velocity C_0 is related to the peripheral and radial components by $\tan \alpha_0 = C_{m_0}/C_{u_0}$ where

$$C_{m_0} = \frac{Q}{2\pi r_0 B_0} \quad \text{and} \quad C_{u_0} = C_{u_1}\frac{r_1}{r_0}$$

B_0 being the width of the guide vanes. The angle α_0 is then determined for each discharge Q from the inlet diagram of the runner (fig. 3.15.) The angle α_0 very nearly coincides with the angular movement of the guide-vane trunnion from its closed position. The guide vanes are linked to the regulating ring for control by the governor as shown in fig. 3.19.

The action of the guide vanes is not a simple throttling action on the water admitted to the runner. By their rotation, the angle α_0 under which the water leaves the guide vanes is varied from the fully open

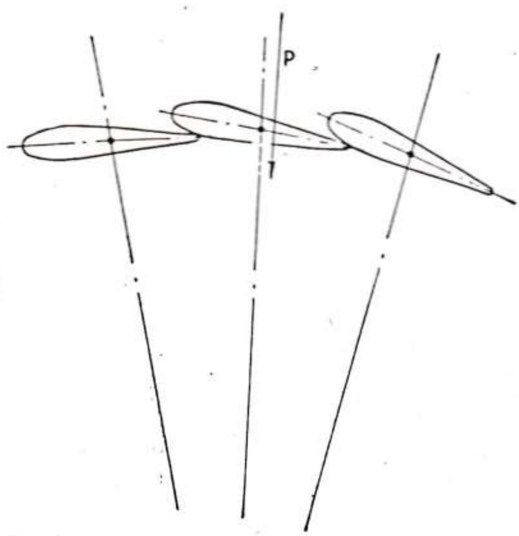


Fig. 3.17.—Section through guide-vane apparatus, closed position

position to the closed position in such a way that as α_0 becomes smaller, the value of $C_{u_1}r_1$ of the vortex at the inlet of the runner increases. The increased centrifugal force acting on the water causes a reduction in pressure at the runner inlet, which in turn reduces the quantity discharged through the runner. Fig. 3.15 shows the variation of the component C_{u_1} at different guide-vane openings, brought about by the change of angle α_1 of the approaching water.

The forces that have to be overcome by the governor vary with each angular position of the guide vanes as illustrated in figs. 3.16, 3.17, and 3.18, where the resultant hydraulic force on one vane is given by P for three different positions. These forces can be calculated from flow lines and pressure diagrams, or they can be measured on model test turbines.

It will be noted that in fig. 3.17, with the guide vanes in the closed position, the force P passes downstream of the guide-vane centre and is directed radially inwards. In fig. 3.16, in the half-open position the force P passes upstream of the guide-vane centre and is inclined relative to its position in fig. 3.17. In fig. 3.18, when open beyond full gate, the force P assumes a very different direction, because the velocity of approach is more tangential than the velocity at exit of the guide vanes. Such a position of the guide vanes is seldom reached. The figure is drawn to show the range of variation of the force P.

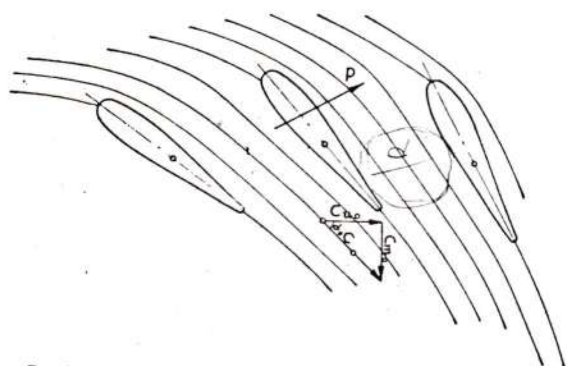


Fig. 3.18.—Section through guide-vane apparatus, open beyond normal position

The pivoting centre of the guide vane is chosen to give best all-round balance of the hydraulic forces over the useful range of openings. Nevertheless, the variation in moment to be overcome by the governor in order to operate the guide vanes is very considerable, as can be seen in fig. 3.20. The maximum moment on the guide vane occurs in the closed position at point A, and this determines the energy required from the governor, after adding suitable allowances for all the frictional resistance in the regulating gear.

Generally, with a well-designed and correctly erected guide-vane apparatus, the governor has to overcome only small resistances during the major part of its closing stroke. At point A and near to the closed position a toggle action of the linkage is utilized to overcome the maximum torque resulting from the hydraulic forces.

At points B and D the guide vanes are in balance. At C the maximum force is required for opening. It will be noted that from D to B the vanes tend to close by themselves. This helps to promote the fast closure of the governor which is necessary to fulfil the regulation performances usually required.

As the closed position is approached, the resistance to closure increases, reaching a maximum at A the fully closed position. This makes the best use of the inertia in the governor gear and at the same time prevents slamming the guide vanes into the closed position.

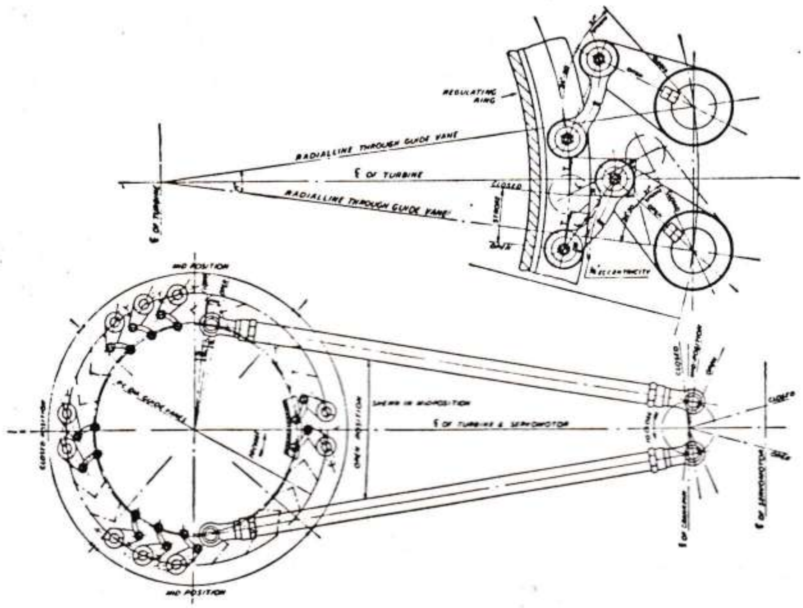


Fig. 3.19.—Linkage between regulating shaft and guide vanes

The toggle action referred to above is illustrated in fig. 3.19. The link which connects the guide-vane lever to the regulating ring is subjected to a compressive force L. Of the two components R (radial) and T (tangential) of L, only T has to be overcome by the governor. The radial component R is taken by the regulating ring working in compression. In the closed position the ratio T/L is least.

4.2. *Protection from Damage.*

Foreign matter carried to the turbine by the water can lodge between guide vanes and prevent their closure. Since all the guide vanes are operated by one governor, an obstruction between two guide vanes will cause the whole governor force to concentrate on those two vanes. To avoid the serious damage that would result, the links which connect the guide-vane levers to the regulating ring are designed as easily replaceable members calculated to fracture when an abnormally high

force is applied to the lever. It is customary to design the breaking member to fracture at three times the normal operating force.

Various devices are used by different manufacturers. Some use notched links which fracture in bending as shown in fig. 3.19; others use links that are made to fracture in shear, whilst some prefer to shear a pin suitably placed between the two parts of a double lever.

By reference to fig. 3.20 it is clear that a guide vane which becomes loose because of fracture of a breaking link or pin will assume various

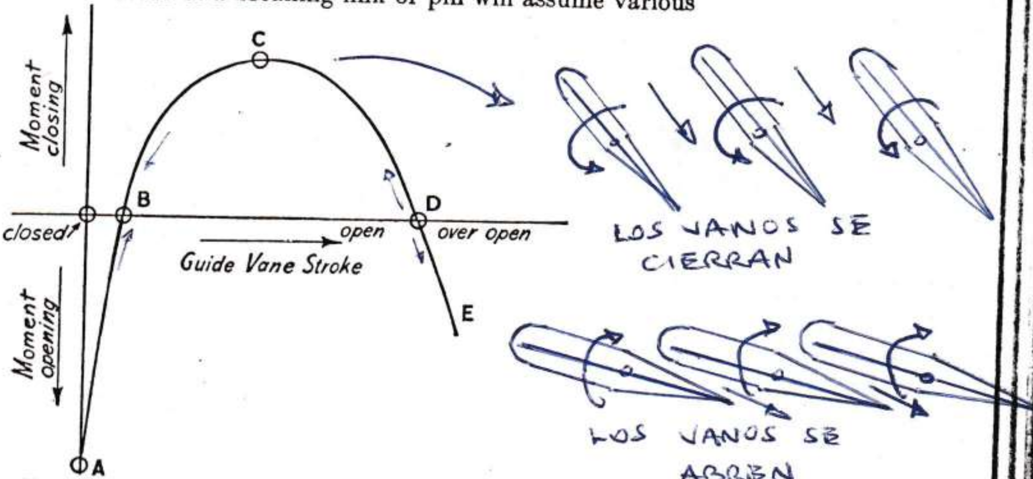


Fig. 3.20.—Variation in the moment of hydraulic force P at guide-vane trunnion

positions according to the gate opening prevailing at the moment of fracture. Hence, if a fracture occurs at any place between A and D, the loose guide vane tends to take up the position B, nearly closed. If fracture takes place between D and E, the vane tends to be thrown wide open and would exceed position E if its stroke were not limited. Stops are normally provided to prevent travel of the guide vanes beyond their normally closed and open positions. However, it must be remembered that fig. 3.20 represents the hydraulic torque on the vanes when all the vanes are at the same angle. As soon as one vane is displaced from the others, the torque on it may change considerably. This has led to oscillation of loose vanes, and it is now common on large machines to provide means of damping the vane movement following fracture of the breaking link or pin, thus preventing damage to the closed and open position stops.

4.3. *Settings.*

The linkage between guide vanes and regulating ring is usually provided with means to permit individual setting of the guide vanes;

this is necessary to ensure that all are equally tight in the fully closed position. It is done by making one of the pins in the link eccentric, so that by its rotation the vane can be moved within small limits sufficient for individual setting in the closed position.

In the tight-closed condition, the governor must not exert its full force on the vanes or else premature breakage of the links will follow. Therefore, it is essential to tighten the guide vanes just enough to take up all clearances when the servomotor is at the end of its closing stroke. The stop limiting the stroke must be in the servomotor itself and not at the guide vanes or other part of the regulating mechanism.

The number of guide vanes varies, according to the size of the machine, from 8 for very small turbines to as many as 28 or more for large machines. They are usually cast in mild steel or fabricated from mild-steel plates. They may even be forged solid from steel billets.

In special cases where severe erosion by sand is to be expected, the guide vanes and the adjacent parts are protected by stainless steel, or the guide vanes can even be made in solid stainless steel. The turbine cover and the pivot ring are often provided with renewable liners when the condition of the water calls for it (figs. 3.3, 3.5, 3.6). It is necessary to make the clearances at top and bottom of guide vanes the smallest practicable in order to minimize the leakage, especially when the guide vanes are closed. This is important because bringing a hydro-electric set to standstill requires application of the brakes to prevent the creeping round at low speed which is detrimental to the thrust bearing. The smaller the leakage past the closed guide vanes, the less the wear on the brake shoes; this is of particular importance when no inlet valve is provided at the turbine. When it is desired to keep the pipeline full during shut-down of the turbine, the leakage becomes doubly important on account of the water wasted. To improve the tightness the guide vanes can be provided with rubber seating strips along the contact surfaces.

In the most widely used designs the guide vane is made integral with its stem, which permits all linkage to be placed out of the water. In fig. 3.8, it will be noticed that the packing ring or seal S_1 is placed underneath the trunnion bearing. This ensures that the water does not wash away the lubricant.

For horizontal-shaft arrangements, as in fig. 3.5, both bearing bushes may be arranged outside the water, using two seals. With vertical-shaft arrangements the bush B_2 is often provided with a seal S_2 to retain the grease in the bearing (fig. 3.8).

5. Spiral Casing.

5.1. *Construction.*

For low heads, up to about 70 ft., and for vertical-shaft units, spiral casings are formed of concrete, without steel-plate lining, generally as shown in fig. 1.4 of Chap. I. The spiral casings are then designed as part of the power-house structure and responsibility for their strength rests with the civil engineer. The turbine manufacturer usually specifies the form of the spiral and supplies the speed ring and stay vanes. Close collaboration between the civil-engineering and turbine designers is essential in order to decide suitable arrangements for anchoring the stay vanes to the foundations, as well as for the attachment of reinforcing bars of the spiral to the speed ring.

For medium heads, the spiral casings are made of rolled-steel plates welded together, and welded or riveted to the speed ring, which in turn can be either a steel casting or of welded fabricated construction (figs. 3.6 and 3.8). For high heads, similar constructions are in use, but due to the very heavy sections required on the speed ring, this item is commonly a steel casting (fig. 3.3). Low-alloy steels of $1\frac{1}{2}\%$ manganese type are often used for spiral casings and for both cast and fabricated speed rings. The higher tensile strength of such materials enables the thicknesses of the spiral plate to be reduced, thus facilitating the welding. High-tensile steels of the quenched and tempered type have also been used for spiral casings and will no doubt become increasingly common as the size of machines increases.

5.2. *Dimensions of Spiral Casing.*

The spiral-casing dimensions are such that the water velocities are approximately:

$$0{\cdot}14\sqrt{(2gH)} \text{ for a low specific speed,}$$
$$0{\cdot}20\sqrt{(2gH)} \text{ for a high specific speed.}$$

In addition, the velocities should not exceed 30 ft./sec. at maximum flow.
9,15 m/s

The inlet to the spiral casing may be chosen equal to or smaller than the diameter of the pipeline, but it is never larger. If a valve is built at the turbine inlet, cost may determine the valve size and hence the spiral inlet diameter. Generally there should be no deceleration of the water as it flows from the pipeline into the spiral casing, and the sections of passage from pipeline to turbine runner inlet must follow this rule in order to minimize the formation of eddies that would

reduce the efficiency. The form of the upstream edge of the stay vanes must also be carefully designed to avoid separation of the flow from the back of the vane. The form of the downstream edge of the stay vane is also important, as choice of incorrect shape can lead to the formation of von Kármán vortices, leading to periodic hydraulic forces which may possibly resonate with the speed ring, particularly on large low-head machines, leading to failure from fatigue.

6. Draft Tube.

6.1. *Hydraulic Design.*

The term "draft tube" designates all water conduits from the exit of the runner to the tailrace where the water has a free level at atmospheric pressure.

Veloci
cero

In reaction water turbines the runner has no contact with the atmosphere and can be placed higher than the tailwater level without losing head, while the absolute pressure at runner exit is proportionally lower. The difference in level between runner exit and tailwater level is called the static suction head H_s (fig. 3.21). Under static conditions, the absolute pressure at runner exit is lower than atmospheric by the water column H_s, provided no air is allowed to enter the draft tube. Moreover, the draft tube has the important function of recovering as much as possible of the velocity head in the water as it leaves the runner. This recovery takes the form of a further reduction in absolute pressure which is expressed by

$\frac{P_2}{\gamma} + H_s =$

$\frac{P_2}{\gamma} =$

$$\eta_D \frac{C_2^2}{2g}$$

called the *dynamic suction head*, where η_D is the coefficient of recovery or efficiency of the draft tube and C_2 is the absolute velocity of water as it leaves the runner. The velocity head $C_2^2/(2g)$ varies in importance according to the specific speed of the turbine. For low specific speeds $C_2^2/(2g)$ is of the order of 3 to 4 per cent of the head H, whilst it reaches 15 per cent or more for higher specific speeds. The recovery of the velocity head is, therefore, very important at high specific speeds, and great attention must then be given to the correct form of the draft tube in order to ensure a good recovery coefficient η_D. The higher the coefficient η_D the smaller will be the exit losses and (for the optimum balance of losses (*a*) and (*b*) discussed in § 3) the lower can be the value of $D_2^3 n/Q_1$ which fixes the size of the runner outlet diameter D_2 and thus has a direct bearing on the cost of the turbine.

In order to prevent cavitation, the turbine must not be placed higher above tailwater level than is permitted by the relation

$$H_s \leqslant H_B - \sigma_c H$$

as shown in Chap. I, § 6. As σ_c depends partly on the dynamic suction head $\eta_D C_2^2/(2g)$, reductions in diameter D_2 lead either to lowering of the turbine and deeper excavation, or to a lesser overall efficiency of the machine, resulting from a smaller η_D.

The recovery of velocity head can be obtained only by gradual increase in the cross-sectional area of the draft tube, resulting in a gradual reduction of the velocity. Too rapid increase of sectional area

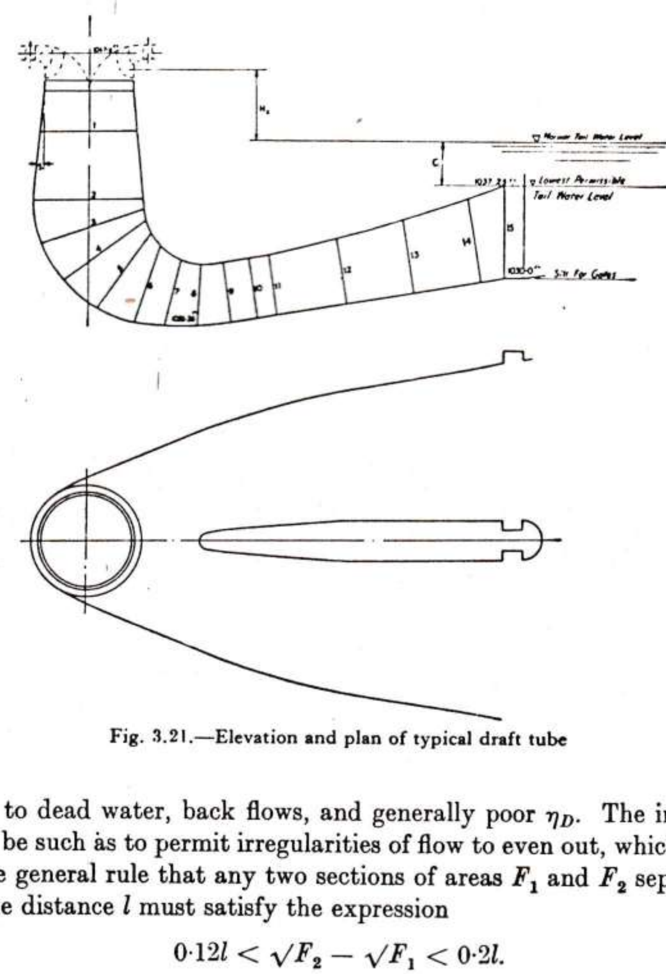


Fig. 3.21.—Elevation and plan of typical draft tube

leads to dead water, back flows, and generally poor η_D. The increase must be such as to permit irregularities of flow to even out, which leads to the general rule that any two sections of areas F_1 and F_2 separated by the distance l must satisfy the expression

$$0{\cdot}12l < \sqrt{F_2} - \sqrt{F_1} < 0{\cdot}2l.$$

On a straight cone of circular section this relation leads to the angle δ formed by the generatrix of the cone and the centre line being $4° < \delta < 8°$. This relation applies generally to sections of the draft

tube that are in a straight line. At the bend, because of the change in direction and the strong tendency for the water to depart from the inner radius, the increase in sectional area is usually made smaller in the first half of the bend, and *nil*, or even negative, in the second half.

For small-size vertical-shaft turbines, the simplest form of draft tube is in a straight cone, discharging sufficiently low under the tail-water level to maintain an ample seal against ingress of air.

This simple arrangement is not possible with horizontal-shaft turbines where a bend must be inserted almost immediately after the runner exit. This bend leads to poor recovery of the velocity head $C_2^2/(2g)$ because of the necessity for placing it close to the runner in order to minimize the space required by it in the power house.

For large-size units, the vertical-shaft setting has become the rule, and the space occupied by the draft tube does not influence the size of the power house, though it often entails deeper local excavation.

In order to keep down the cost of such excavation, particularly in rock, there is a temptation to shorten the vertical leg of the draft tube (between runner exit and bend). This leads to poorer turbine efficiency because a bend too close to the runner causes non-symmetrical flow at the runner exit. It is most desirable to keep the straight part of the cone at least equal to one runner diameter. The capital cost of extra excavation for the draft tube should be compared with the gain in energy output over the lifetime of the installation.

The bend that turns the water from the vertical to the horizontal direction is designed to combine the highest recovery factor η_D with minimum depth of excavation. Considerable variations in form have been used in the past, the latest trend being to avoid complicated structures and to shape the bend with the largest possible inner radius, and with cross-sections spreading sideways. Fig. 3.21 shows in elevation and plan the form of a typical draft tube with high recovery coefficient. Fig. 3.22 gives cross-sections through the same draft tube. The following features are to be noted.

The transition from a circular section in the vertical leg to a rectangular section in the horizontal leg takes place in the bend, which tends to spread the water horizontally to the left and right of the horizontal leg. To prevent excessive spreading, the oval sections from 4 to 8 are so designed as to offer a concave surface to the flow of water. This produces a better distribution of water in the horizontal leg.

The horizontal leg of the draft tube is generally inclined upwards to lead the water gradually to the level of the tailrace and to prevent entry of air from the exit end. In this respect, and having regard to turbulence

being unavoidable at part gate-opening, the water seal or cover C at normal tailwater level should not be less than one-half runner diameter. The lowest tailwater level should not uncover any part of the roof of this horizontal leg (fig. 3.21).

The flow of water in the draft tube is much more regular at loads near Q_1 than at part gate-opening, where the water is discharged with a considerable rotational component of the velocity (fig. 3.15). This leads to irregular flow at the bend of the draft tube.

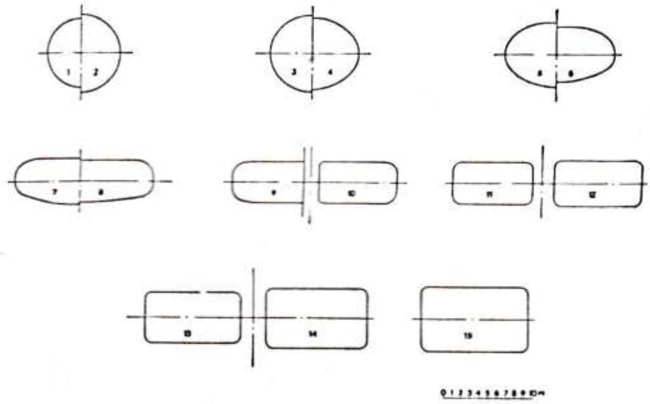


Fig. 3.22.—Sections through draft tube of fig. 3.21

Pulsating-flow conditions develop at part gate-opening which have been found to be produced generally as follows:

Because of small guide-vane opening, the component C_{u_1} is large and results in relatively low pressure at the runner inlet. The runner is not " saturated " and, because of the large centrifugal forces, the flow occupies the space at the runner exit indicated by (*) in fig. 3.23. The flow in this space (*) is subject to important tangential components of velocity *in the direction of rotation of the runner* forming a vortex in the positive direction. Low pressure results in the middle of the draft tube partly occupied by dead water indicated by (m). This dead water is being lifted towards the runner. A train of vortices occur between the vortex (*) and the dead water (m) as indicated in fig. 3.23. At a somewhat larger guide-vane opening (but still below half opening) the dead water space (m) is reduced and the train of vortices gradually join in a single vortex, the core of which is indicated in broken lines. This core is eccentric to the draft-tube axis, unstable in space and subject to a definite precession movement. The core of the vortex has

the appearance of a swinging rope. This precession gives rise to cyclic pressure changes at the rate of one every 4 to 6 runner revolutions.

Periodical changes in the form of flow thus take place, producing periodical pulsation in magnitude of the driving torque of the turbine which, under certain conditions, can be observed as pulsations in the power output of the generators.† To remedy this state of affairs, air can be introduced in the draft tube at part guide-vane opening. The air

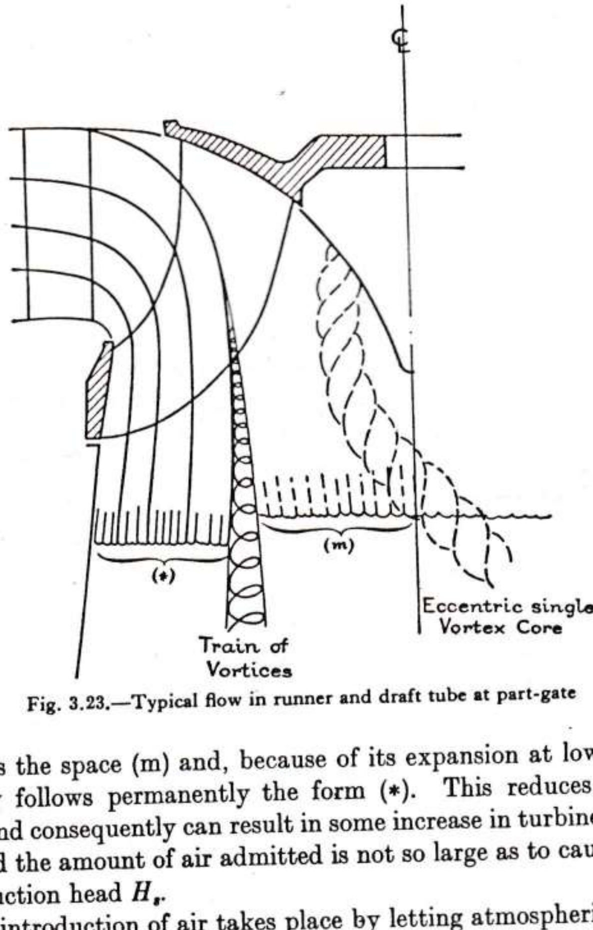


Fig. 3.23.—Typical flow in runner and draft tube at part-gate

then fills the space (m) and, because of its expansion at low pressure, the flow follows permanently the form (*). This reduces losses by eddies and consequently can result in some increase in turbine efficiency provided the amount of air admitted is not so large as to cause a fall in static suction head H_s.

The introduction of air takes place by letting atmospheric air enter the draft tube through an automatic air valve. Its control is usually

† Pulsation in power output is not always caused by the flow in the draft tube. The periodical swing observed on the kW.-meter may be caused by electrical unsteadiness in the transmission system and in such cases it cannot be cured by the introduction of air into the draft tube.

mechanical, causing the air valve to open on sudden closure of the guide-vane apparatus, and to remain open from about 0·5 or 0·6 gate-opening to the fully-closed position. At higher guide-vane openings, air is not desirable; if introduced in large quantities, it would reduce the efficiency by losing suction head. The effect of introducing air is particularly noticeable at no load, when its presence reduces the churning of water by the runner and, therefore, the turbine reaches normal speed at a substantially smaller guide-vane opening.

6.2. *Construction.*

For large draft-tube dimensions the strength of the power-house structure requires a pier or supporting wall placed in the middle of the horizontal leg, as shown in fig. 3.21. This pier is of no assistance towards efficiency of the draft tube, but it facilitates the power-house construction by the added support of the draft-tube roof. At the same time it reduces the span of the stop logs or gates which are required at the draft-tube exit to permit dewatering for inspection. This pier is usually protected at its upstream end by a steel nose piece. The draft tube is lined with steel plates where velocities exceed about 15 ft./sec. The vertical cone, almost invariably, is steel-lined. Eddies at the runner exit, which occur particularly at changes of loads or at small guide-vane opening, necessitate careful anchorage and thorough grouting of the liners in the foundations. Pressure-grouting through holes specially provided through the plates is used to fill any air pockets. Tie bars or anchor irons are welded to the stiffener ribs of the steel liners and to the nose piece on the central wall.

7. Shafts.

Turbine shafts are normally forged, but for the largest sizes fabricated shafts have been used. The stresses allowed for transmission of the driving torque to the generator are kept low, so that not more than normal stresses are reached under the heaviest acceleration that can take place, namely during short-circuit conditions of the generator. It is usual to choose the shaft diameter $d \simeq 4(N/n)^{1/3}$ in. where d = shaft diameter, N = turbine output in b.h.p., n = speed in r.p.m. The combined rotating element of generator and turbine is checked for shaft critical speed.

The turbine shaft can be subdivided if the distance to the generator and the handling by the crane make it necessary. The shaft is bolted to the turbine runner and to the generator shaft by rigid coupling flanges. The torque is transmitted either by fitted bolts or by a transverse key, which is usually in two parts so that the keyway need not be

cut across the shaft spigot. After manufacture, the shaft sections are carefully checked for truth individually and again after assembly with turbine runner and generator shaft. Owing to the very large weights involved, this check is best done in the vertical position, as all deflections are then eliminated.

It is usual to bore the shafts with a central hole, for the purpose of inspection of the forging. Flaws that usually occur in the central portion of the forging are then eliminated and perfect soundness of the shaft can be ascertained by visual inspection. Test pieces are taken from the forging, usually at each end, and in longitudinal and tangential positions.

8. Shaft Gland.

Where the shaft emerges from the turbine cover, a suitable sealing arrangement is fitted to prevent water escaping or air entering. Different designs are in use, each presenting its own advantages and limitations:

(*a*) Throw rings and labyrinths can be used successfully wherever the tailrace water level is sufficiently low. The advantage of this design is the avoidance of any packing and, therefore, no maintenance is required and no wear takes place. Such an arrangement is shown in fig. 3.5. It must be observed that, when filling the spiral casing before starting up, the air in the turbine is blown out through this type of gland and may entrain some moisture in the bearing nearby. Careful priming with release of air through suitable piping is therefore the rule.

(*b*) Another type of stuffing box comprising two large rings of graphite-impregnated asbestos packing separated by a metallic lantern ring and pressed by a gland ring with grease lubrication, provides a tighter seal. Maintenance is necessary. The gland must be loose enough to permit some leakage of water, which is necessary for cooling. If over-tight the gland will heat up. If the water is silt-laden wear will soon occur. The shaft must then be fitted with a renewable liner, which necessitates shut-down for replacements.

(*c*) Fig. 3.3 shows still another type of gland which consists of three segmented carbon rings, held close to the shaft by garter springs, separated by a metallic lantern ring. This design requires less maintenance than type (*b*) but is more difficult to fit up properly. The water must be free of sand, or else the gland box must be fed with clean water from an external supply, so that the pressure in the box is higher than in the turbine, and entry of sand is prevented. A renewable sleeve is normally fitted to the shaft.

9. Hydraulic Thrust.

The resultant of all hydraulic forces on the runner and its weight must be borne by the bearings. Where the turbine is of vertical-shaft setting, the rotating parts of the generator and turbine comprise the predominant load on the thrust bearing. It is then unnecessary to introduce design complications in order to secure insignificant reduction of the hydraulic thrust. In the case of a horizontal-shaft arrangement,

the thrust bearing takes only the hydraulic thrust and it is then of importance to reduce it to a minimum. The hydraulic thrust is calculated as the resultant of the following forces:

(*a*) *Hydraulic pressure on the area between runner seals.* This can be reduced to nil if the seals are made at the same diameter on both sides of the runner inlet.

(*b*) *Hydraulic pressure on the runner crown.* The water can be assumed to rotate in the space between runner and cover at half the angular velocity of the runner. The pressure distribution varies as the square of the radius and thus the load is comparable to the weight of a paraboloid of rotation. This load can be lessened to a certain extent by preventing the rotation of the water in the space between runner and cover, and at tne same time draining it away.

(*c*) The water leaking through the seals must be evacuated to the tailrace (fig. 3.5), or to the draft tube through the runner as shown in figs. 3.3, 3.6 and 3.8. A certain amount of *back pressure* at the runner crown can be eliminated by suitable positioning of the discharge passage but can seldom be completely suppressed. When pipes are fitted to discharge into the tailrace the back pressure at the runner crown is reduced to the friction losses in these pipes.

(*d*) As the shaft passes through the gland, it is subjected to *an axial force caused by the difference in pressure between both sides of the gland*, one side at draft-tube pressure, the other at atmospheric pressure.

(*e*) The water changes its direction inside the runner; from a radially inward one, it is turned into an axial direction. *The change in the momentum of the water imparts a force to the runner which is proportional to the product of the discharge and the change of axial component of velocity.* This force is therefore proportional to the square of the discharge and is directed towards the generator, i.e. in the direction opposite to that of the water leaving the runner.

The resulting axial thrust is the algebraic sum of the forces obtained under items (*a*) to (*e*). It will be appreciated that most of these components vary with the discharge, while others vary with the speed of rotation of the runner.

For a horizontal-shaft turbine, the thrust to be taken by the bearing may not always be in the same direction at all loads, and it is therefore necessary to restrain the shaft in both directions.

It is desirable to avoid reversal of thrust because of the likelihood of causing vibrations at the point of reversal. The runner seals and pressure chambers on each side of the runner are designed to produce a one-way thrust, usually in the direction of the discharge, away from the generator. Nevertheless, limitation of the shaft movement in both directions is essential.

10. Bearing and Lubrication.

If all hydraulic pressures were symmetrical around the runner only the weight of the rotor would have to be accounted for by the journal

bearings of a horizontal-shaft arrangement, and no side thrust would develop with the vertical-shaft setting. The inlet pressure is not, however, wholly symmetrical all round because of the friction losses in the spiral casing which leads the water to the guide apparatus. On the other hand, the runner and shaft must be restrained against possible vibrations set up by irregularities of flow and against the tendency of the runner to move out of central position. For these reasons a very substantial guide bearing is provided on the turbine cover shown in figs. 3.3, 3.6, 3.8.

For dimensioning the bearing it is convenient, and practical experience has proved it satisfactory and sufficient, to assume a side thrust equal to one-tenth of the hydraulic force that would result if the full head were acting on one side of the runner, over the width B_0 of the guide vanes. The cooling requirements for the bearing should be determined by calculating the power required to shear the bearing oil film, assuming a uniform oil film. This will normally amount to about one thousandth of the rated output of the machine. This heat will have to be eliminated by the oil circulation. The oil can be circulated by a pump through a conventional cooler with cooling coils submerged in water. Alternatively, water is passed through cooling coils or in jackets around the bearing (fig. 3.3).

Almost without exception the bearings are lined with Babbitt metal and oil-lubricated. The lubricating pumps can be driven by gears from the main shaft or by separate electric motors. Alternatively, the guide bearing for vertical-shaft setting can be designed for self-lubrication where the oil circulation is induced by the shaft rotation without requiring any external pump. For horizontal-shaft bearings, oil rings can provide for lubrication of the journal bearing and cooling coils can dispose of the heat, or else the oil is circulated by pumping through a cooler or coils.

In many cases the cooling water can be tapped conveniently from the spiral casing and discharged into the tailrace. A strainer is provided which can be cleaned without interruption of service, and a multiple diaphragm reduces the pressure and controls the quantity of cooling water.

For heads exceeding about 300 ft., cooling water taken from the pressure supply entails wastage of energy. In these cases, low-pressure water can be circulated from the tailrace by electric-motor-driven pumps. This introduces some complication in the equipment of the power house and the need for a standby provision of cooling water,

either by taking a supply from the pipeline, or by duplication of the pump and motor. To obviate these, cooling coils are sometimes arranged in the tailrace or in a convenient part of the draft tube. Ejectors can also be used with advantage to circulate low-pressure water.

11. Selection of Speed, Runner Level, and Turbine Dimensions.

It is helpful to have a basis for rough estimation of the principal dimensions of the turbine, which will enable a projected power house to be laid out. The dimensions arrived at, however, are subject to adjustment by the turbine and generator manufacturers when the project is further advanced.

11.1. *Choice of Speed.*

In Chap. I, § 5, the procedure for selecting the type of turbine is explained and fig. 1.2 gives the highest specific speed compatible with the available head.

For convenience we reproduce in fig. 3.24 that part of fig. 1.2 which concerns Francis turbines.

It is assumed that the output N per machine has been chosen. From the definition of the specific speed n_s the highest permissible speed is calculated as

$$n = n_s \frac{H^{5/4}}{\sqrt{N}}$$

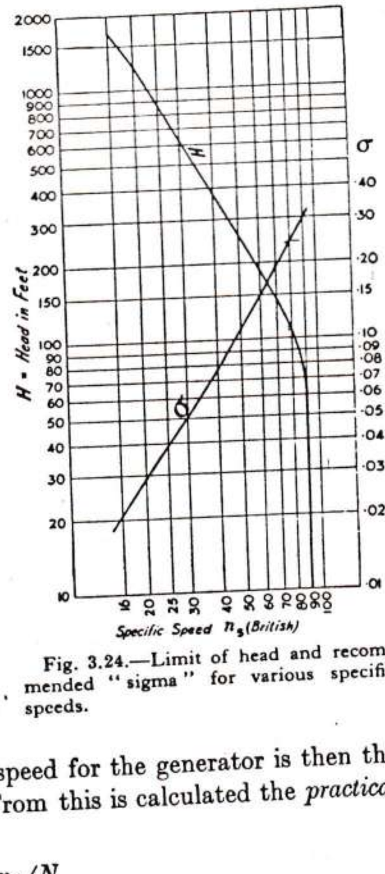


Fig. 3.24.—Limit of head and recommended "sigma" for various specific speeds.

where n_s is the highest permissible specific speed for H.

The nearest lower synchronous speed for the generator is then the highest *practical speed of the set.* From this is calculated the *practical* value

$$n_s = \frac{n\sqrt{N}}{H^{5/4}}$$

11.2. *Runner Level.*

Fig. 1.5 of Chap. I gives for the chosen n_s the corresponding σ. The highest safe value of H_s is then calculated from

$$H_s \leqslant H_B - \sigma H$$

where H_B is the barometric water column as obtained from fig. 1.6. For convenience fig. 3.24 shows the values of σ given in fig. 1.5.

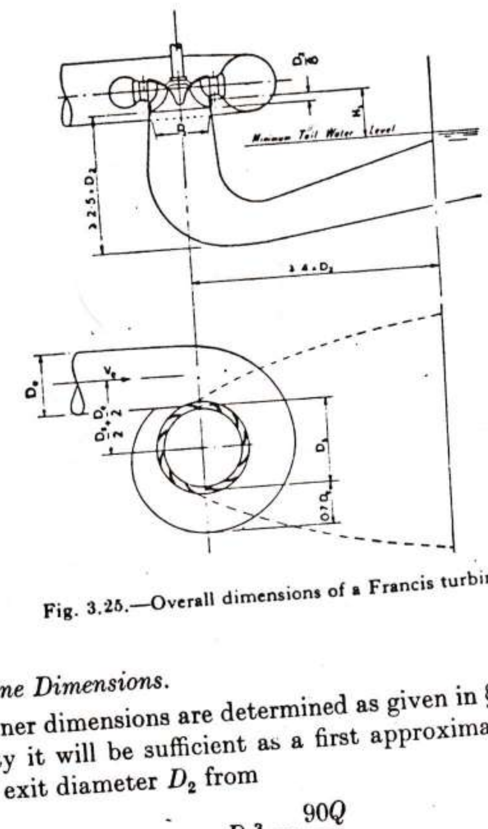


Fig. 3.25.—Overall dimensions of a Francis turbine

11.3. *Turbine Dimensions.*

The runner dimensions are determined as given in § 3. For the sake of simplicity it will be sufficient as a first approximation to calculate the runner exit diameter D_2 from

$$D_2{}^3 = \frac{90Q}{n}$$

where Q is the discharge at full load in cusecs and n the speed in r.p.m.

From D_2 follow the overall dimensions of the draft tube. From fig. 3.25 the depth measured from runner exit D_2 must be at least equal to $2{\cdot}5D_2$. A greater depth is preferable but the cost of the foundations is increased.

The length of the horizontal leg of the draft tube should be at least four times D_2, and the area of exit such as to reduce the water exit velocity to 4 to 5 ft./sec.

The spiral-casing entrance diameter D_e is calculated from § 5 by assuming v_e to be 0·14 to 0·20 $\sqrt{(2gH)}$.

The mean diameter of the stay vanes D_s can be taken approximately as

$$D_s = 1{\cdot}5 D_{1\,\text{exterior}}$$

$D_{1\,\text{exterior}}$ is calculated from the coefficients given in fig. 3.11, making the spiral entrance tangent to the circle D_s (fig. 3.25). The overall dimension of the spiral casing is then approximately

$$1{\cdot}7 D_e + 1{\cdot}5 D_{1\,\text{exterior}} \text{ (see fig. 3.25).}$$

The height of the gate apparatus B_0 is obtained from the ratio $B_0/D_{1\,\text{exterior}}$ as read from fig. 3.11.

11.4. *Example.*

A head of $H = 96$ ft. is available, and it is proposed to instal one Francis turbine of $N = 26{,}000$ b.h.p. at an altitude of 1600 ft. above sea-level. From fig. 3.24 read opposite head $H = 96$ ft. the maximum permissible value of n_s is 85.

$$n = 85\,\frac{96^{5/4}}{\sqrt{N}} = 85 \times \frac{300}{161} = 159 \text{ r.p.m.}$$

The nearest lower synchronous speed is $n = 150$ r.p.m.
The specific speed for this speed (150 r.p.m.) is

$$n_s = \frac{150\sqrt{N}}{H^{5/4}} = \frac{150 \times 161}{300} = 80{\cdot}5.$$

From fig. 3.24 for $n_s = 80{\cdot}5$, $\sigma = 0{\cdot}24$.
From fig. 1.6, $H_B = 31$ ft. (for altitude 1600 ft.).
The runner level must not be higher than

$$H_s \leqslant H_B - \sigma H = 31 - 0{\cdot}24 \times 96 = 8 \text{ ft.}$$

above the minimum tailwater level.

Turbine dimensions.

Assuming an efficiency $\eta = 90$ per cent the discharge at full load is

$$Q = \frac{550N}{\gamma H \eta} = \frac{550 \times 26{,}000}{62{\cdot}4 \times 96 \times 0{\cdot}90} = 2650 \text{ cusecs.}$$

The runner exit diameter is approximately

$$D_4 = \sqrt[3]{\left(90\,\frac{Q}{n}\right)} = \sqrt[3]{\left(90\,\frac{2650}{150}\right)} = 11{\cdot}66 \text{ ft.}$$

D_2

The draft-tube depth below runner exit is approximately $2{\cdot}5 \times 11{\cdot}66 = 29$ ft.
The horizontal leg is $4 \times 11{\cdot}66 \simeq 47$ ft.

The area of exit $= \dfrac{2650}{5} = 530$ sq. ft.

From $H = 96$ ft.. $\sqrt{(2gH)} = 78{\cdot}4$ ft./sec. and the coefficient ϕ for $D_{1\,\mathrm{exterior}}$ is read from fig. 3.11 for $n_s = 80{\cdot}5$ as $\phi_1 = 1{\cdot}13$.

$$D_{1\,\mathrm{exterior}} = \frac{60\phi_1\sqrt{(2gH)}}{\pi n} = \frac{60 \times 1{\cdot}13 \times 78{\cdot}4}{\pi \times 150} = 11{\cdot}33 \text{ ft.}$$

$$v_e = 0{\cdot}20\sqrt{(2gH)} = 0{\cdot}20 \times 78{\cdot}4 = 15{\cdot}8 \text{ ft./sec.}$$

The area of inlet pipe is $\frac{1}{4}\pi D_e^{\,2} = \dfrac{Q}{v_e} = \dfrac{2650}{15{\cdot}8} = 168$ sq. ft.

$D_e = 14{\cdot}6$ ft. diameter.
The overall dimension of the spiral is approx.

$$1{\cdot}7D_e + 1{\cdot}5\,D_{1\,\mathrm{exterior}} = 1{\cdot}7 \times 14{\cdot}6 + 1{\cdot}5 \times 11{\cdot}33 = 41{\cdot}8 \text{ ft.}$$

From fig. 3.11 for $n_s = 80{\cdot}5$

$$\frac{B_0}{D_{1\,\mathrm{exterior}}} = 0{\cdot}32, \text{ therefore } B_0 = 0{\cdot}32 \times 11{\cdot}33 = 3{\cdot}62 \text{ ft.}$$

The profile of the runner is completed by reading from fig. 3.11

$$\phi = 0{\cdot}79 \text{ for } D_{1\,\mathrm{interior}} = 7{\cdot}9 \text{ ft.}$$
$$\phi = 0{\cdot}38 \text{ for } D_{2\,\mathrm{interior}} = 3{\cdot}8 \text{ ft.}$$

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CHAPTER IV

Kaplan and Propeller Turbines

REVISED BY G. T. KEAST

ORIGINAL AUTHORS G. H. LUNDGREN AND C. E. STILLE

1. General Characteristics.

Turbine designers from 1920 onwards have been directing their attention to units of higher specific speeds in order to reduce machinery cost in low-head hydro-electric power stations. The Francis turbine, with a specific speed up to approximately 400 per runner, was at that time sufficiently advanced to give excellent results for most low-head schemes. For low heads generally, where the machinery costs are an appreciable proportion of the total expenditure, any increase in specific speed leads to a greater reduction in cost per horse-power than would the same increase for higher heads. In consequence the advent of Kaplan and propeller turbines, which are high-specific-speed machines, has enabled hitherto uneconomic hydro-electric projects to be developed.

As a result of these efforts the propeller turbine emerged. Experiments by Professor Kaplan with a model propeller turbine led him to the conclusion that the great disadvantage which it had of a very peaky efficiency curve could be overcome if the pitch of the blades could be altered during normal operation as the output changed. The variable-pitch propeller turbine, more appropriately called the Kaplan turbine, has now reached a stage of evolution where, with few exceptions, it can be used with marked advantage over its predecessor, the fixed-blade or propeller turbine. As development and experience increased, the limiting head for the Kaplan turbine was gradually raised until the present general value of approximately 50 m. for all outputs has been reached (1968). Smaller Kaplan turbines can be used for heads as high as 70 m. The limiting head for propeller turbines is approximately 36 m.

Fig. 4.1 gives a section of a typical medium-head Kaplan turbine of moderate dimensions with a plate-steel spiral casing. It will be seen that, with the exception of the runner and its associated parts, the

general mechanical design is similar to that of Francis turbines (Chap. III). The height of the guide vanes is, however, somewhat greater in proportion to the runner diameter.

After forty years of operating experience, the added complication of the variable-pitch blades of Kaplan turbines no longer presents any serious problems in commissioning or maintenance, although of course the blade-operating mechanism and its associated oil hydraulic control system inevitably increases the cost of the machine.

The design varies with the maximum operating head and physical dimensions. For small outputs and low heads the type shown in fig. 4.2 generally satisfies requirements. Fig. 4.3 shows a typical older design for larger outputs and low heads. At approximately 10 m. head details of design must be modified, as indicated in fig. 4.4, to meet the higher water pressure. At heads of approximately 20 m. the stay-ring design can be altered to facilitate erection without much increase in cost, and an example of this for a small turbine is shown in fig. 4.5. Although the design with independent stays can be used for heads greater than 25 m. it is seldom justified with small units where concrete shuttering would be difficult.

A plate-steel spiral casing is generally adopted for heads exceeding 23–32 m., the former being the limit for the smaller units (figs. 4.1, 4.6 and 4.7). It should be noted, however, that where very large units are adopted and the quality of the concrete is in doubt, it is advisable to resort to a plate-steel spiral for heads in excess of approximately 27 m.

Very cheap low-head units for small outputs may be equipped with inside regulation of the guide vanes, where the linkage gear for movement of the guide vanes is immersed in water, as shown in fig. 4.2. This construction is subject to the effect of permanent immersion in water, and it is difficult to repair the damage if an object is wedged between two guide vanes and the turbine shut down. With this contruction the guide vanes pivot on fixed bolts holding the top and bottom covers together and, from the hydraulic point of view, it has the advantage that there are no losses due to separate stay vanes. Outside regulation, as used for the other turbines shown, is more usual and enables a larger proportion of the weight of the generator, hydraulic thrust, and power-station structure to be taken through the stay vanes to the foundations.

The propeller turbine runner (with fixed blades) consists essentially of the hub, the blades, and the nose piece. For the smallest sizes the runner can be a one-piece casting, although in most cases the nose piece

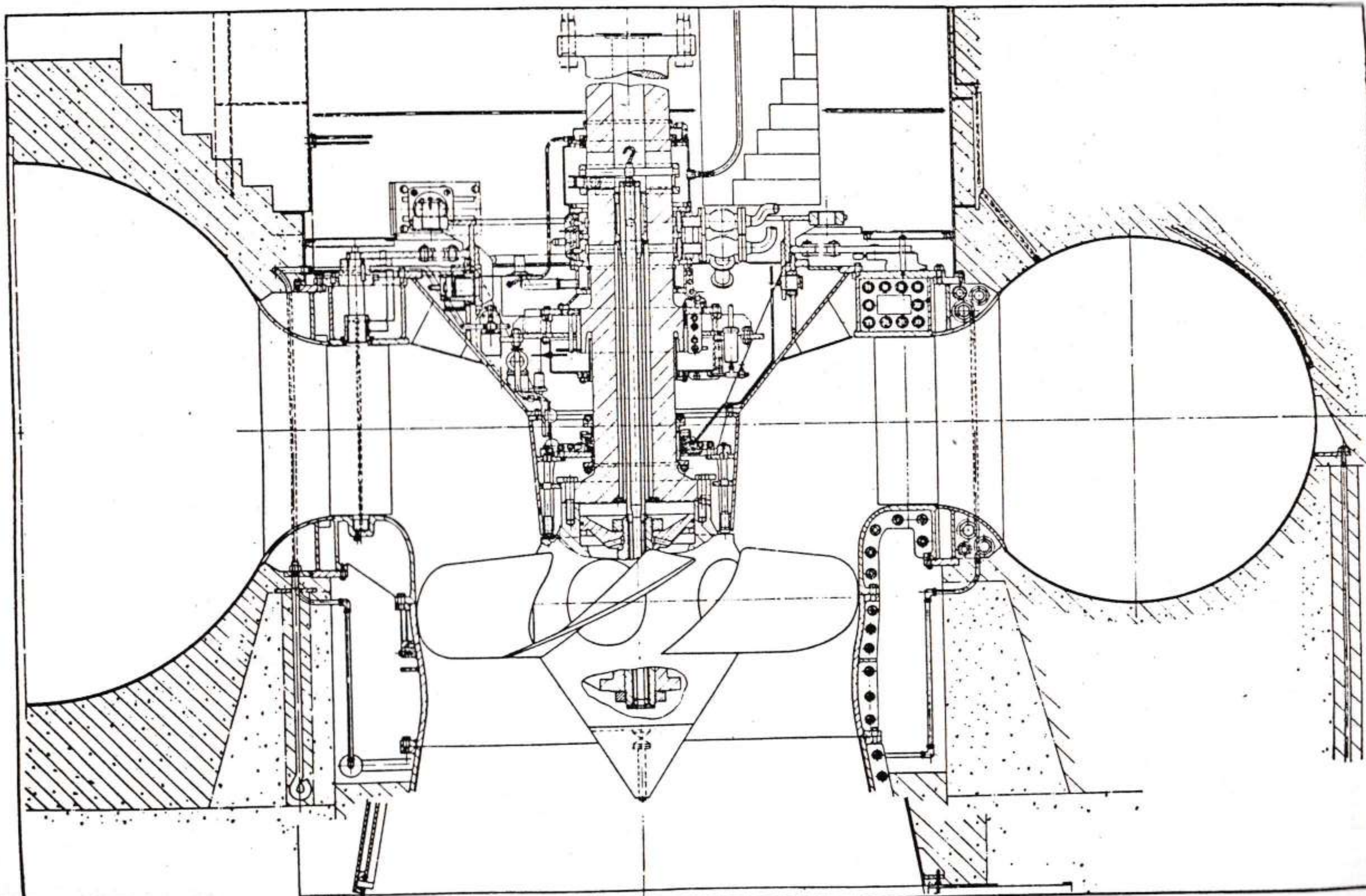

Fig. 4.1.—Kindaruma turbine (Kenya)
Output 20.9 MW. Head 35–40m. Speed 214.3 r.p.m.
Maximum Runner Diameter 3.25 m

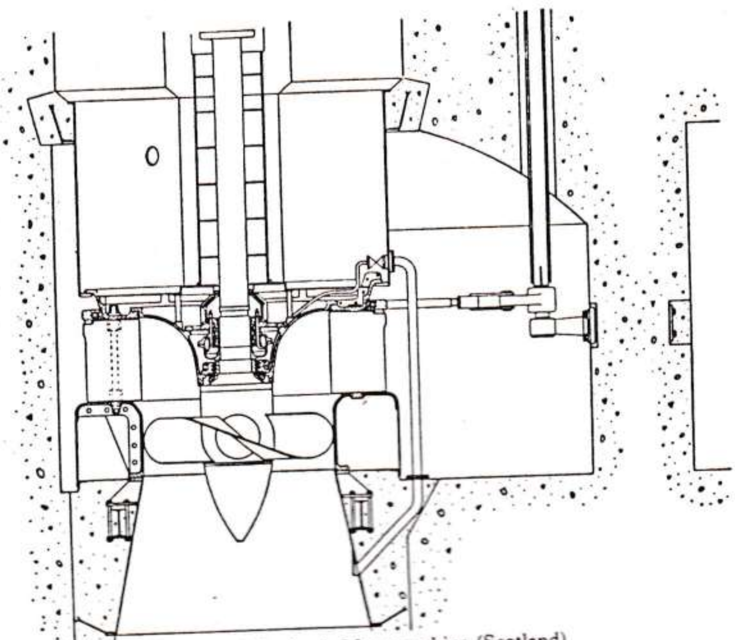

Fig. 4.2.—Loch Morar turbine (Scotland)
Output 420 kW. Head 5.8–7.9 m. Speed 333 r.p.m.
Maximum Runner Diameter 1.3 m

Boving

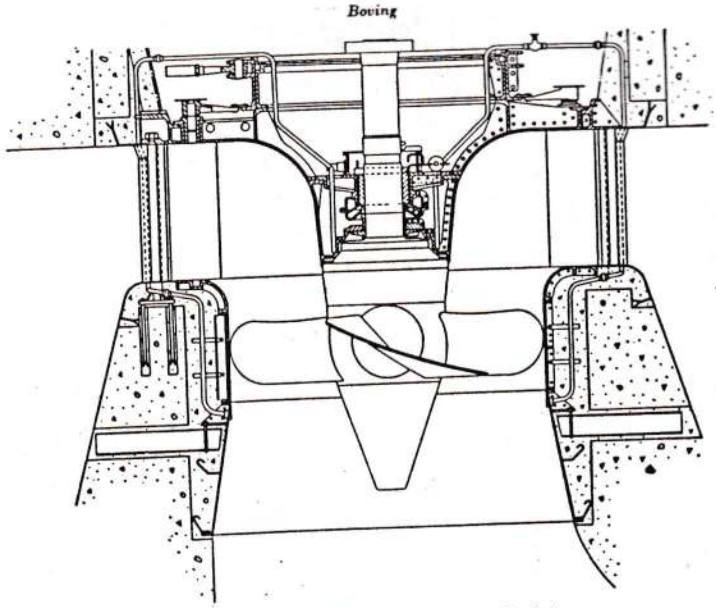

Fig. 4.3 —Muhammedpur turbine (India)
Output 3 16 MW Head 5.35m. Speed 125 r.p m.
Maximum Runner Diameter 3.6 m

Boving

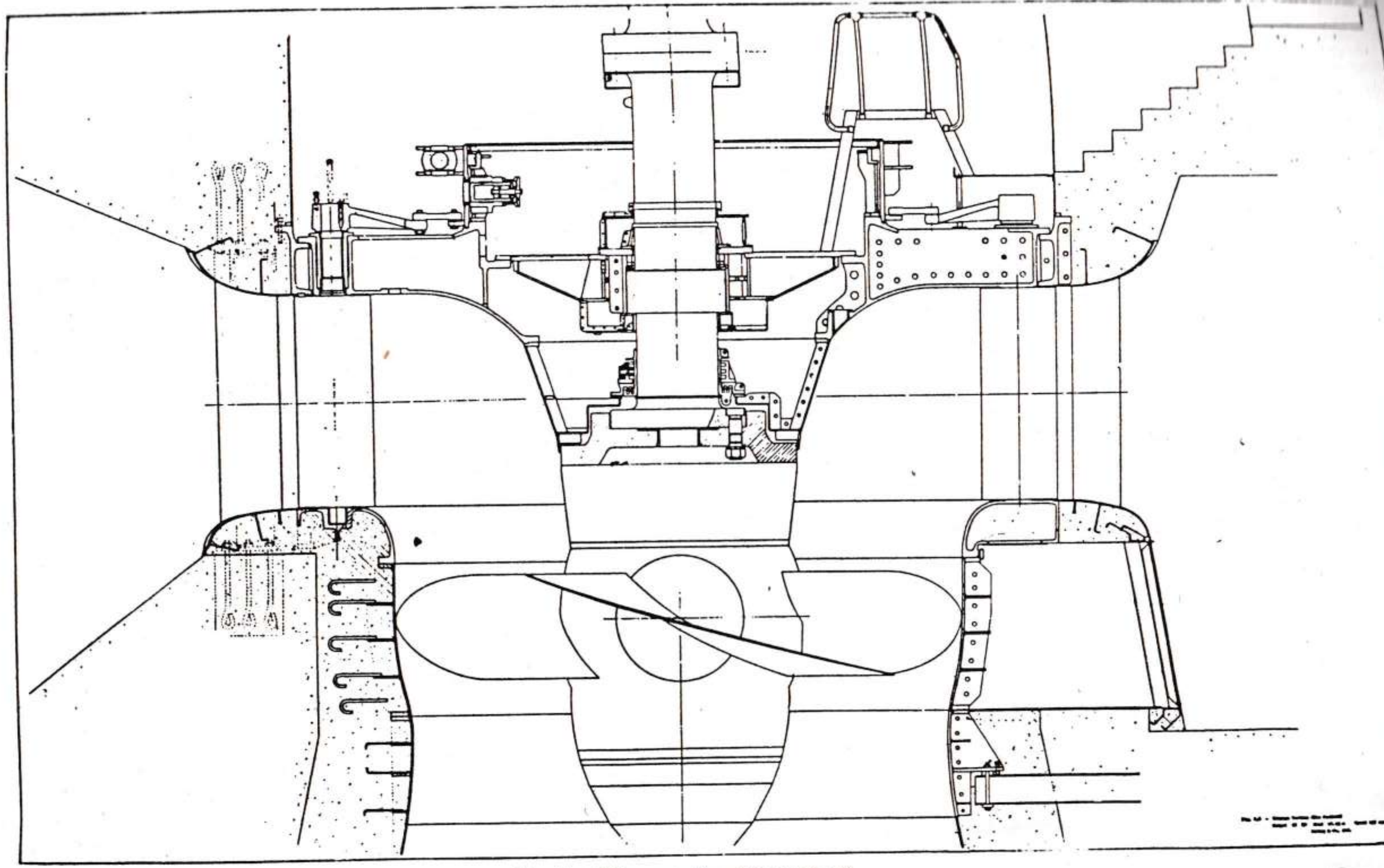

Fig. 4.4.—Waipapa turbine (New Zealand)
Output 18 MW. Head 16.15 m. Speed 125 r.p.m.
Maximum Runner Diameter 4.88 m

Boving

is separate and of cast iron and is bolted to the hub. Although when cast in one piece runners can be as large as transport limitations allow, the use of a propeller or Kaplan turbine has some advantages over the Francis type in this respect, in that the blades and hub can be made separate, thus enabling units of very large dimensions to be used. The limiting factors are then the weight and dimensions of the hub. The lower cost per horse-power of large units and their higher efficiency may, in many cases, more than compensate for the extra cost of detachable blades.

The Kaplan turbine runner is hydraulically similar to the propeller

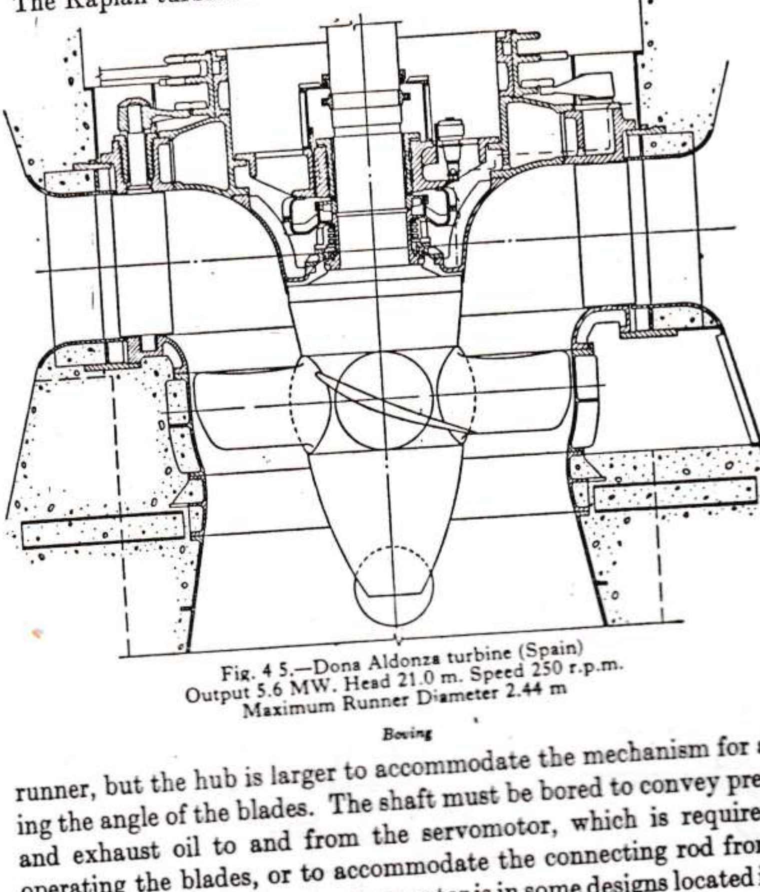

Fig. 4 5.—Dona Aldonza turbine (Spain)
Output 5.6 MW. Head 21.0 m. Speed 250 r.p.m.
Maximum Runner Diameter 2.44 m

Boving

runner, but the hub is larger to accommodate the mechanism for altering the angle of the blades. The shaft must be bored to convey pressure and exhaust oil to and from the servomotor, which is required for operating the blades, or to accommodate the connecting rod from the servomotor. The oil-pressure servomotor is in some designs located inside the runner hub; in others it is housed in a bulge in either the turbine or generator shaft. The servomotor is equipped with a control mechanism or combinator (p. 159) to ensure that the correct relation between

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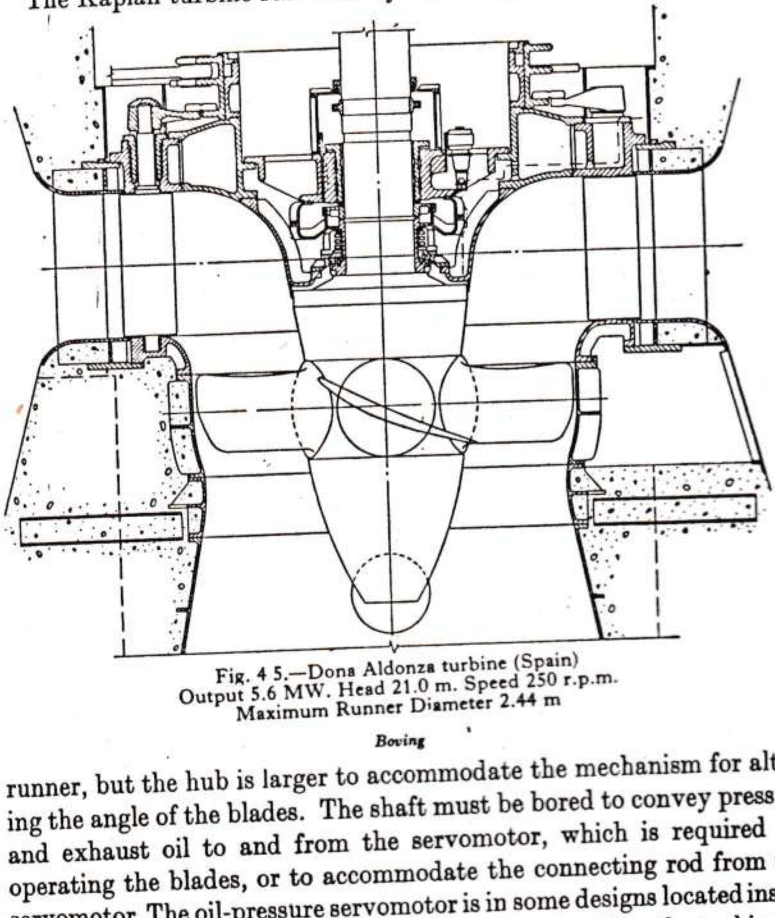

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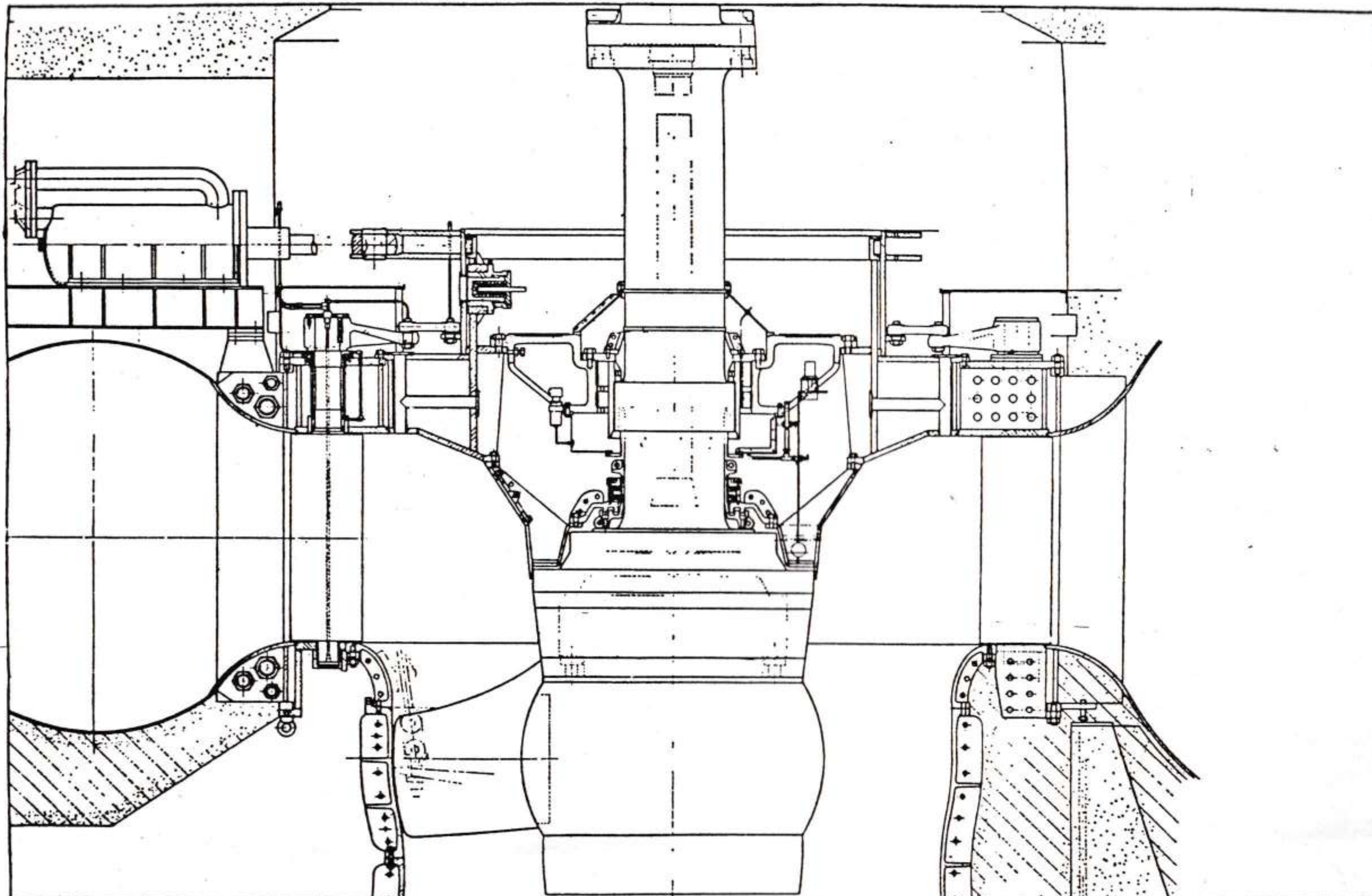

Fig. 4.6.—Inverawe turbine (Scotland)
Output 26 MW. Head 29.25 m. Speed 166.7 r.p.m.
Maximum Runner Diameter 4.29 m

Boving

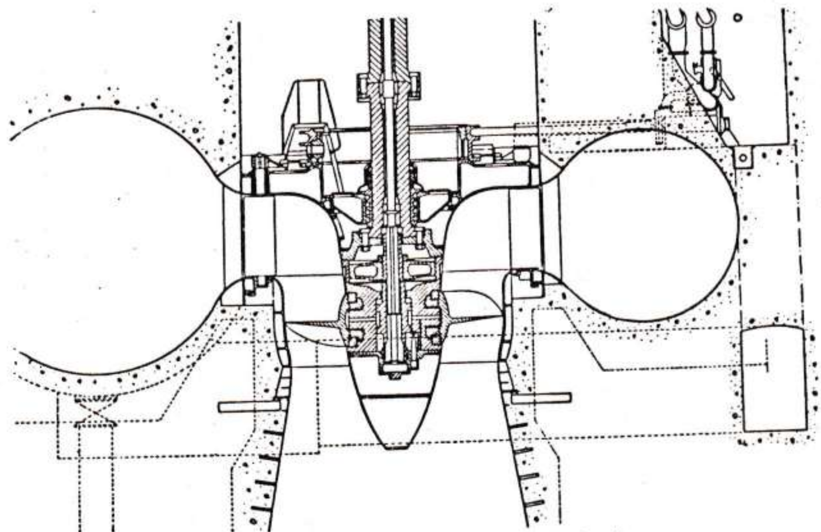

Fig. 4.7.—Karapiro turbine (New Zealand)
Output 31.5 MW. Head 30.5 m. Speed 166.7 r.p.m.
Maximum Runner Diameter 4.35 m

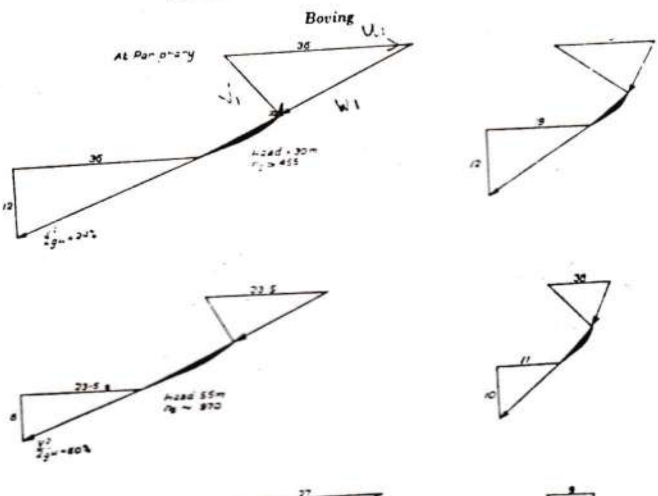

Fig. 4.8.—Nominal velocity diagram

guide-vane opening and blade pitch is automatically obtained throughout the operating range of the turbine. A mechanical connection to this control mechanism from either the guide-vane servomotor or the governor must be provided in addition to the necessary oil-piping.

The outstanding feature of both propeller and Kaplan turbines is the high peripheral velocity of the blades in relation to the spouting velocity $\sqrt{(2gH)}$.

Fig. 4.8 shows the water velocity, both at the hub and the periphery, for three turbine runners of different specific speeds. It will be noted that the axial velocity increases in relation to the peripheral speed as the specific speed decreases, and that the velocity of water over the blade surfaces increases as the operating head is raised. It will also be noted that the kinetic energy remaining in the water as it leaves the runner, expressed as a percentage of the net head, increases with specific speed.

2. Turbine Runners.

2.1. *Efficiency*.

In fig. 4.9 some efficiency curves are shown for a model Kaplan turbine tested at various running speeds, the results having been transposed to apply to a model runner of 1 m. diameter operating under a constant head of 1 m. It will be seen that the relatively flat efficiency curve is in fact the envelope of the best operating points of an infinite number of propeller turbines with different fixed blade angles. For each propeller curve there is one optimum combination of guide-vane opening for its particular blade angle, and at any other guide-vane opening the performance of the machine falls away steeply, often accompanied by rough running and cavitation inception.

This illustrates the importance of maintaining the correct relationship between guide-vane opening and blade angle over the whole range of output of the Kaplan machine, which is the primary function of the combinator mechanism described on p. 159. It also indicates the advantages of a Kaplan turbine in that for the same maximum efficiency (occurring at zero-degree blade angle) it has a much greater overload capacity than the corresponding propeller turbine or, conversely, for the same maximum output, the efficiency of the moving-blade Kaplan is much higher over most of the load range.

At very low loads ideal combination is of less importance to efficiency, but operation without vibration is facilitated by keeping the angle of attack low, i.e. by maintaining as far as possible the lowest blade angle compatible with the small guide-vane openings.

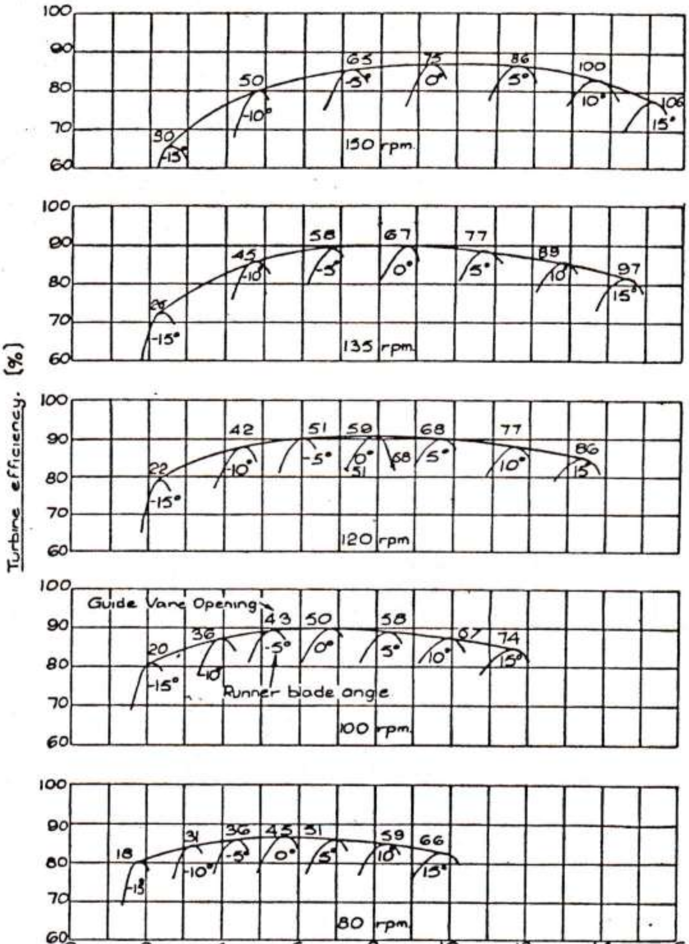

Fig. 4.9.—Model runner efficiency curves

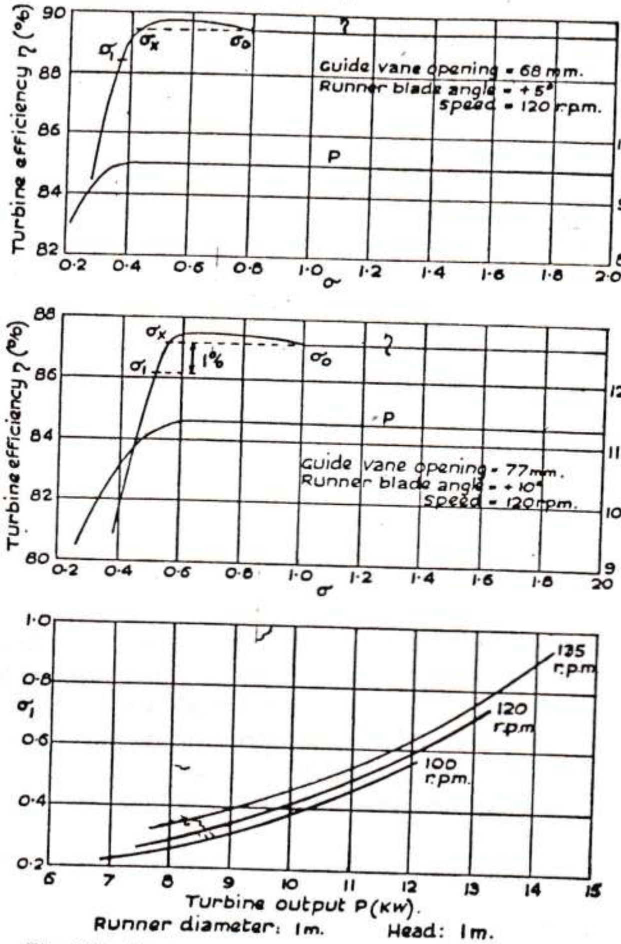

Fig. 4.10.—Typical Kaplan Turbine Cavitation Characteristics

$$\sigma = \frac{H_B - H_s}{H}$$

$$\sigma_c = \frac{H_B - H_c}{H}$$

As fig. 4.9 shows, the optimum combination or blade-angle/guide-vane-opening relationship varies for different unit speeds, which means that for any given size of turbine running at a fixed synchronous speed, the combination must be adjusted to suit the prevailing net head if the best envelope of efficiency is to be maintained over the load range. The reason for this will be evident from an examination of the velocity diagrams for the leading edge of the blade. Thus where a turbine has to operate over a considerable range of head, provision must be made in the combinator to permit such adjustment.

The choice of a high speed, i.e. small diameter, and consequently

lower machinery cost, necessitates a small positive or even a negative suction head, and deep excavation for the draft tube is therefore necessary. If the specific speed conforms to normal practice and the necessary depth of excavation for a normal runner design is greater than desirable, a different runner design should be chosen which for the same output gives a lower sigma value (see Chap. I, § 6). This results in a design with a greater number of blades and/or blades of a lower aspect ratio.

2.2. *Cavitation.*

When the velocity of water over a convex curved surface is sufficiently high the local pressure at certain points will drop to a value below that of the vapour pressure, resulting in vapour-filled cavities. The formation of such cavities on the blade surfaces leads to an alteration in the hydraulic characteristics of a turbine. For any condition of speed or output, a Kaplan or propeller turbine has a safe limit to the suction head H_s, i.e. the distance between runner and tailwater level up to which operation will be free from cavitation. The effect of raising the runner relative to tailwater level will be to reduce the pressure on all parts of the blades, and in areas where the pressure is already low the formation of cavities may take place. If the turbine is to operate at high altitudes the local pressure in some part of the runner may, because of the reduced barometric pressure H_B, be low enough to cause cavitation which would not normally occur.

For further information on this subject the reader is referred to Chap. I, § 6 and Chap. III, § 3.

The value of the critical cavitation factor σ_c changes with output and peripheral speed, i.e. with specific speed for any runner type, and it is important for satisfactory operation that it is not exceeded by overloading the turbine.

Cavitation may also occur at the clearance between the runner blades and chamber; this necessitates careful dimensioning of the clearance in relation to the blade thickness. The phenomenon is of lesser importance as regards reduction of output or efficiency but can result in undesirable pitting of the runner chamber and blade edges unless the more cavitation-resistant types of material are used.

The upper two curves of fig. 4.10 show the variation in efficiency and output caused by a gradual lowering of the tailwater level relative to the runner centreline (i.e. a gradual reduction of σ). For this illustration the same model turbine as for fig. 4.9 has been used and the corresponding operating points, at optimum combination, can be

found on the envelope efficiency curve for the unit speed of 120 r.p.m. The lower curve, corresponding to a higher turbine output, shows how cavitation affects performance at a much higher tailwater level, indicating the need for a deeper runner submergence when high specific outputs are required.

At the reduced tailwater level appropriate to σ_0 the degree of cavity formation is such as to begin to affect the efficiency but not deleteriously. The slight increase in efficiency which is often shown at this operating point has been attributed to the reduction in viscous drag at the blade tips associated with the development of clearance cavitation. At still lower tailwater levels (appropriate to σ_x) the performance of the machine begins to fall away as flow separation occurs at the inlet edges with the resultant disturbance of the flow regime over the blade profile. As the tailwater level is progressively lowered still further, both output and efficiency fall steeply as flow separation and bubble formation become more extreme, even to the extent of reducing the discharge area by choking the passages between the blades.

arrastre visco

The lowest curve of fig. 4.10 shows the variation of σ_1 with turbine output at various unit speeds, where σ_1 is defined as that value of σ for which the turbine efficiency is reduced by 1% below its value under non-cavitating conditions.

Owing to the unstable nature of the cavities formed due to low local pressure, cavitation is accompanied by vibration to varying degrees. With high heads cavitation may result in severe vibration even on the power-house building. Where vibration is marked and the safe sigma value is not exceeded, it may be that the combination, i.e. relation between guide-vane opening and blade angle, is not correct. In some cases, usually at light loads, an alteration of the combination may eliminate rough running.

The contribution of research into the cause and nature of pitting is very great but it has not yet been possible to use such information for making definite design limits beyond which it would be inevitable that pitting would occur in any particular plant. There is, however, much experience which enables the designer to be confident that, if a certain specific speed is exceeded for the given hydraulic conditions, pitting is liable to occur, and due consideration must be given to this when specifying or selecting the speed of a Kaplan or propeller turbine.

It should be appreciated that when less-resistant materials are used, pitting can occur even if the plant σ-value is safe on the basis of careful laboratory tests. With stainless-steel blades finished to close limits,

particularly at the leading edge, the possibility of pitting is quite small even for heads as high as 30 m. At lower heads the likelihood of pitting with the same material is less, provided a normal value of specific speed is chosen and a well-tried hydraulic design adopted.

2.3. *Some Hydraulic Characteristics.*

Fig. 4.11 gives the relation between runaway speed, water quantity, blade angle and guide-vane opening. Two curves of runaway speed are superimposed, one being the maximum (off-cam) with the most unfavourable combination and the other with normal (on-cam) combination. The maximum runaway speed for the propeller turbine of this type will be considerably less than for the Kaplan, since the guide-vane opening will correspond to that value required to give maximum output, say 70 with a blade angle of 5°, i.e. about 230 r.p.m., whereas with a Kaplan turbine it would be about 250 r.p.m. due to the lower blade angles permitted by the different combination. If the head variation is large, and/or should the head for which the best efficiency is required be low in relation to the maximum head, the overspeed ratio, i.e. maximum runaway speed over normal speed, will be higher than for normal conditions with only a small head variation.

Several points regarding the effect which the hydraullic characteristics of Kaplan turbines have on speed and pressure reguation should be noted. A quick opening movement of the runner blades is required, otherwise unstable governing may result, particularly if one Kaplan turbine is the only source of power in a system. The possibility of using Kaplan turbines for the purpose of frequency control would not exist if means were not provided for opening the runner blades in approximate synchronism with the guide vanes.

Another hydraulic characteristic of Kaplan and propeller turbines is the upward thrust of the runner which occurs with small guide-vane openings and large blade angles. This effect increases with the speed of rotation and is most noticeable during momentary speed-rise conditions when full loads or overloads are rejected. The thrust may be large enough to overcome the weight of the rotating masses. The support for the rotating masses in such circumstances moves from the thrust bearing to the runner centre-line and may cause the rotor of an umbrella-type generator (Chap. VIII, § 6) to throw. To reduce this effect where it is likely to occur, generally when the flywheel effect of the rotor is low, the guide-vane servomotor may be equipped with a special slowing-down device which delays complete closure of the guide

RPM

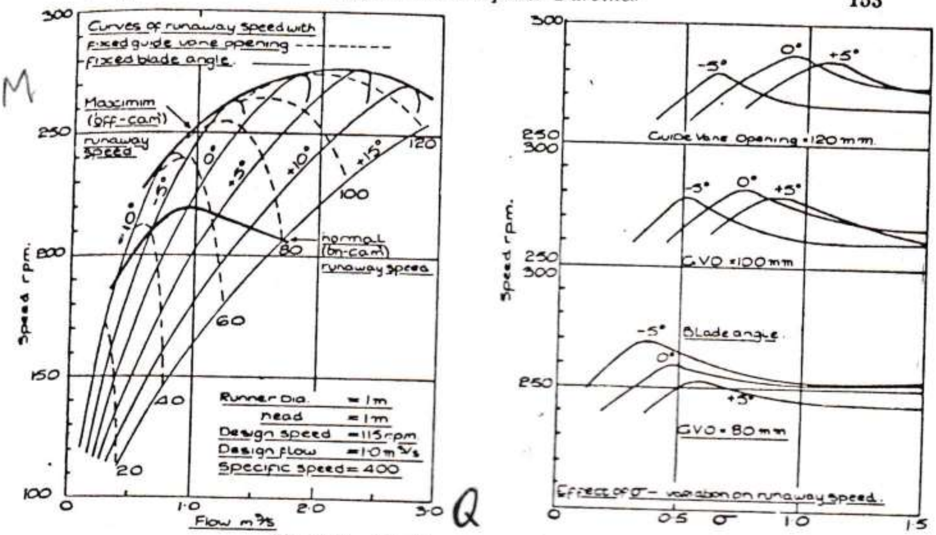

Fig. 4.11.—Model runner performance curves

vanes until the momentary speed-rise has decayed to a value more nearly approaching normal speed. Some manufacturers prefer to install anti-vacuum valves for this purpose which have the same effect, namely to stabilize the transient pressure fluctuations which occur between runner blades and guide vanes under load rejection conditions.

It will be seen from the runaway-speed curve (fig. 4.11) that the guide-vane opening is large at no load if the blade angle is large. Off-load regulation can be improved in two ways without incurring risk of water-column rupture in the draft tube. One method is to use a short closing time of the guide-vane servomotor and admit air to the runner chamber. The other is to use a longer time, sufficient to ensure against draft-tube water-column rupture, and to provide for slow closing of the blades.

At on-cam runaway speed the water quantity can be as much as twice that for full load. It is desirable that any device which could cause the servomotor to close the guide vanes must be accompanied by a sufficiently long closing time to avoid the possibility of disastrous pressure changes. An air-inlet device can only ensure against draft-tube column rupture and not against excessive pipeline pressure rise.

The effect of cavitation must be considered in connection with runaway speed and the appropriate correction made. Fig. 4.11 shows the effect of variation of the cavitation factor on the runaway speed. The flow through the runner increases with the speed, and this has the effect of reducing the pressure on the back of the blades until, when a certain

speed is reached, cavitation commences. A further decrease in the cavitation factor would cause the speed to drop, as the nature of the flow tends to that for an impulse turbine. The runaway speed of the Kaplan turbine requires careful calculation from test data to ensure an accurate value so that a safe, but economical, design of the generator is obtained.

Kaplan and propeller turbines are characterized by a large hydraulic thrust, which is a few per cent less than the gross area of the runner multiplied by the maximum head. The effect of combination is to reduce this thrust load a little as the output is increased, thus giving a maximum value towards no-load.

2.4. *Blades and Hub Design.*

As the head increases, the loading on the blades increases, and a larger hub in relation to the overall diameter of the blades is required in order to accommodate the blade bearings. This conforms with hydraulic requirements since a small hub diameter for a low peripheral speed would necessitate an undesirably high degree of curvature of the blades adjacent to the hub in order to obtain the correct angle of attack.

As the specific speed varies, the shape, aspect ratio, and number of blades are altered to ensure that the shape of the efficiency curve and value of the cavitation factor are suitable for the operating conditions specified. The type of blade for the lowest heads has a relatively high aspect ratio, i.e. the angle subtended at the axis of rotation by the leading and trailing edges is small. When such a runner is viewed along the axis of rotation there will be no overlap of the blades, even in the closed position. Such an arrangement gives a low blade area, but leads to a high value of cavitation factor and prevents the economic use of such a runner at high heads owing to the large negative suction head that would be needed. With the high peripheral speeds required to obtain a high specific speed it is essential to reduce the wetted surface of the blades as much as possible, in addition to altering the shape of the blade sections, so that friction losses are kept low.

As the operating head increases, consideration of the desired suction head and cavitation factor necessitates increased overlap of the blades either by increasing their number or by decreasing the aspect ratio. The interference effect of adjacent blades reduces the suction peak near the leading edge. Furthermore a reduction in peripheral speed is required to obtain an acceptable value of cavitation factor and efficiency. In the higher-head ranges there appears to be a divergence of design as regards number of blades and aspect ratio. It is possible to obtain the same degree of overlap of blades, either by a low aspect ratio and a small

number of blades, or by a higher aspect ratio and a larger number of blades. For cost or performance there does not appear to be any marked advantage with either design.

The necessity of increasing the hub diameter as the head becomes greater reduces the net area of the water passage; the result is that, for any given output and head, the size of a Kaplan turbine at higher heads is larger than that of a Francis turbine and the advantage obtained by a higher specific speed is smaller. At about 20 m. the Kaplan turbine size measured at the pitch-circle diameter of the guide vanes is the same as that of a Francis turbine. For lower heads than this the Kaplan turbine is smaller.

The hub is generally spherical at the blade roots so that the clearance between the hub and blades is constant as the latter rotate. For low heads, however, the hub is sometimes made cylindrical since then the varying clearance between the blades and hub has only a small effect on performance.

2.5. *Construction.*

The propeller runner in its smallest sizes is a single-piece casting, and for low heads cast iron is quite satisfactory. For high heads cast steel is used, and in some cases certain areas of the blade, as shown by experience to be liable to pitting, have cast recesses which are filled by welding with stainless steel.

The largest sizes have separate blades bolted to a cast-steel hub either by a large single nut or by a clamping ring and several bolts. For the highest heads (above 20 m.) separate blades of cast stainless steel can be used.

Typical medium and large Kaplan runners are shown in figs. 4.12 and 4.13. The runner blades are, with a few exceptions for small plants with a low operating head, of cast stainless steel. This material, of low nickel content, has been found to be completely satisfactory. In few plants (ranging in operating head up to 50 m.) has there been any serious pitting requiring the removal of the runner blades for repair by welding.

The water velocity in the vicinity of the blade surface is very high. It is essential, therefore, that the shape is accurate to close limits and that the finish is quite smooth, particularly at the leading edge, the suction or lower side of the blades and over the profiles in the peripheral zone.

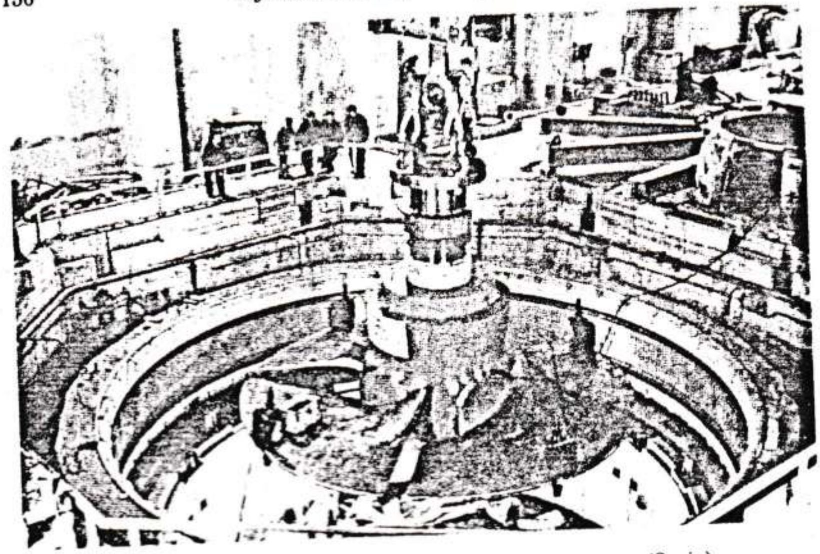

Fig. 4.12.—Installing the Kaplan runner of Velle turbine (Spain)
Output 40·6 MW. Head 14·1 m. Speed 75 r.p.m. Max. runner diameter, 7·1 m.

Boving

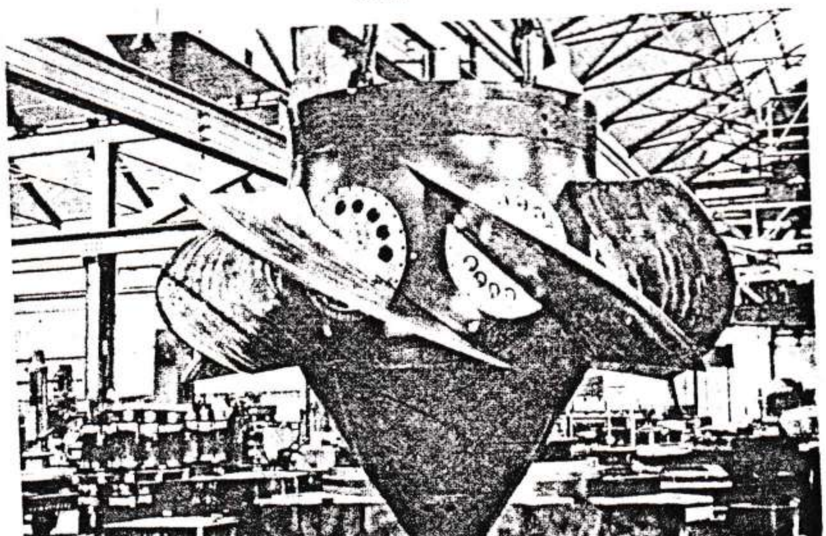

Fig. 4.13.—Runner of Krokstrommen turbine (Sweden)
Output 50 MW. Head 57·7–59·5m. Speed 231 r.p.m. Max. runner diameter, 3·5m.

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Apart from certain of the earliest Kaplan turbines which were provided with regulating gear, operated by hand or an electric motor, almost all have an oil-operated servomotor automatically controlling the blade angle, according to some predetermined relationship with the opening of the guide vanes.

Three distinct designs of mechanism for operating the blades exist:

(*a*) The design known as Englesson's Patent—first used in 1922 and shown in fig. 4.14—has the servomotor located in the hub above the axes of the blades. The basic principle of the conversion of the reciprocating motion of the servomotor piston to an oscillating motion of each blade is achieved by moving a block up and down—the motion being transferred to a trunnion or pin on a ring bolted to each blade. A housing for these blocks, one to each blade, is provided by the piston rod to which the piston is fixed by means of a large nut. A lower and an upper bearing of equal diameter are provided for the piston rod, and above the upper bearing is a seal to prevent oil leakage from the lower side of the servomotor cylinder.

Admission of oil to either the upper or lower side is obtained by means of the hollow combinator-valve rod and connecting piping by which pressure oil is led from the combinator located above the turbine cover. This valve rod, made of copper alloy, is stationary and the cast-iron liner which is bolted to the piston rod rotates outside it. These two items a~e machined and drilled so that movement of the valve rod uncovers ports and grooves, causing the piston rod to follow the valve-rod movement with inherent return motion. The dimensions of the ports and grooves are made to give the required rapid opening movement when the load is increased and slow movement when load is reduced. The liner inside the piston rod is drilled and machined to allow escape of exhaust oil to the outside of the inner tube connected to the valve rod.

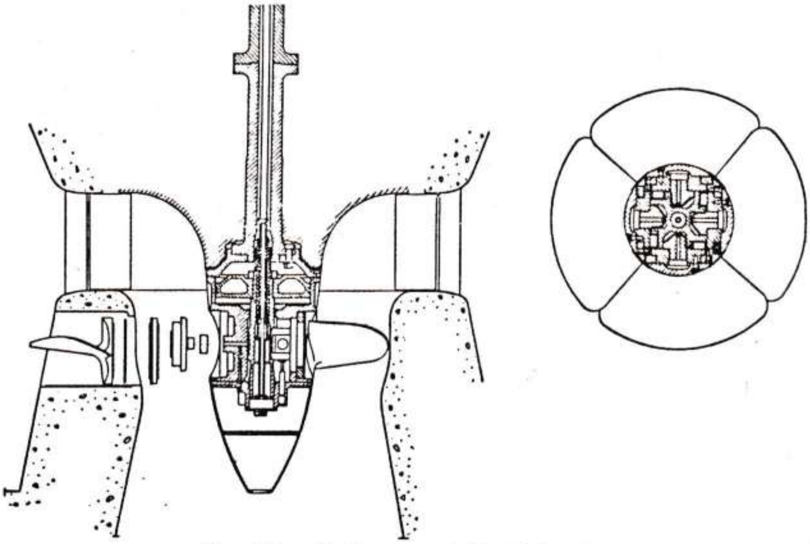

Fig. 4.14.—Kaplan runner hub—Design A

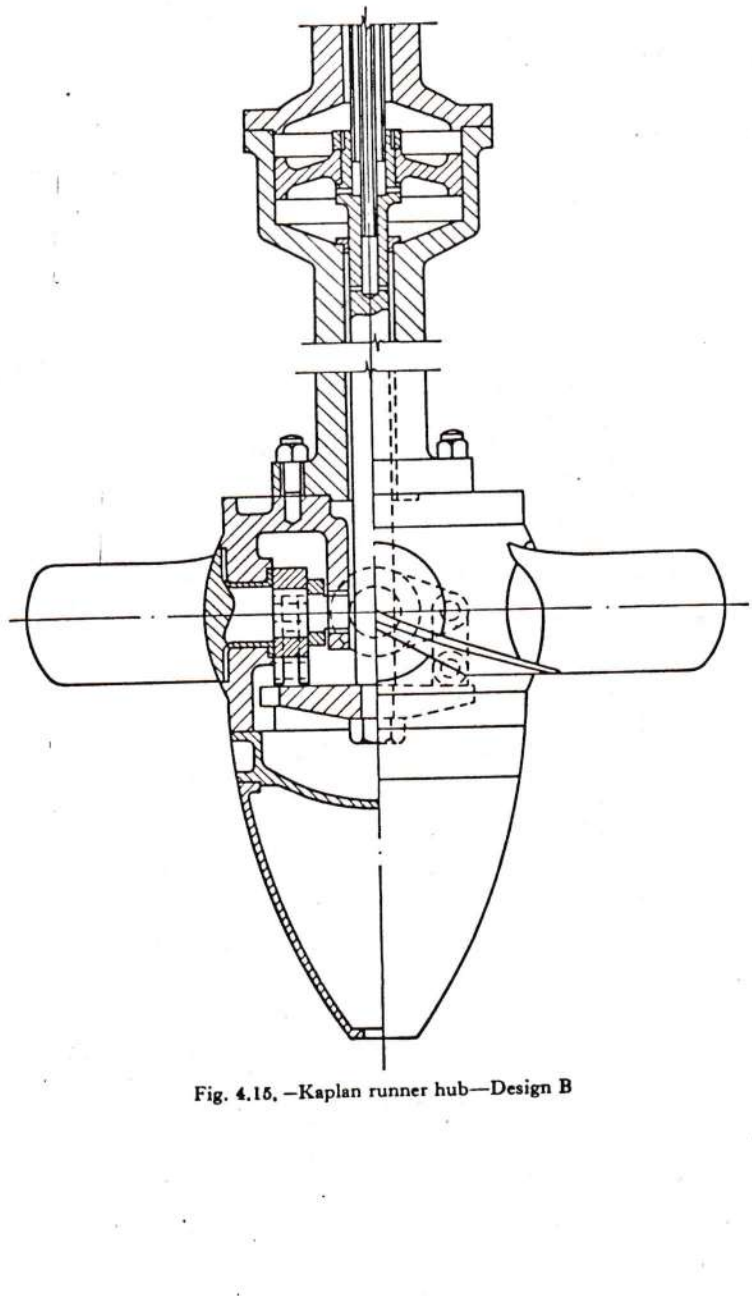

Fig. 4.15. —Kaplan runner hub—Design B

(*b*) Another type, shown in fig. 4.15, has the servomotor located in a bulge in the shaft remote from the runner. The servomotor piston is connected by a rod to the crosshead located in the hub below the turning axis of the runner blades. Each blade has a lever keyed to its shaft and a link connecting the lever to the crosshead. The reciprocating movement of the crosshead thus gives the required turning movement to the blades. The crosshead is prevented from turning relative to the hub by means of a guide. As in design (*a*) the whole blade-operating mechanism operates in an oil bath giving the necessary lubrication to the bearing surfaces.

Three bearing surfaces are provided for the blades, which are in one piece. The vertical load is taken on an outer and inner bearing, with the centrifugal load taken at the face between the outer-bearing bush and the blade lever.

Separate circular covers are generally provided to facilitate the inspection and renewal of the seals between the blades and the hub without removal of the former.

(*c*) A third type, introduced in 1940, has the servomotor located below the runner axis in the nose piece or conical fairing at the bottom of the runner. The piston is stationary and the cylinder itself moves.

The mounting of the blades in their bearings is similar to that used in design (*b*). In design (*c*), however, the lower end of the lever of each blade terminates in a pin, on the inner end of which a block is mounted. This block slides in a slot machined on one of the facets (one for each blade) of the servomotor cylinder. The cylinder is prevented from rotating by projections from it bearing against surfaces provided in the lower part of the hub. As the grooves on the cylinder facets are skewed they impart an oscillatory movement to the blades when the servomotor cylinder moves up and down.

The hub servomotor operates in a similar manner to the first design described and, as with the two other designs, an oil bath is provided for the lubrication of all moving parts.

3. Combinator.

For Kaplan turbines the correct relationship between blade angle and guide-vane opening is maintained automatically over the operating range of the turbine by a device referred to as the *combinator* or runner-blade control mechanism.

It is in essence a runner-blade positioning servo-mechanism identical in principle with the main regulating or distribution valve which governs the movement of the guide-vane servometer. It consists of four principal components:

(1) The main regulating valve with pressure oil supply and oil exhaust ports to and from the runner-blade servomotor.

(2) The auxiliary pilot valve which initiates and controls the movement of the main regulating valve.

(3) The mechanism delivering the input signal from the actuator to the auxiliary pilot valve.

(4) The feedback mechanism from the runner-blade servomotor which restores the equilibrium of the system once any movement has been initiated.

There are so many different arrangements of these four essential components that it is impossible within the space permitted to describe more than a small proportion.

Fig. 4.16 shows in diagrammatic form a typical modern arrangement of an electro-hydraulic combinator mechanism, designed for mounting on top of the generator shaft in line with the most common practice.

Any movement of the actuator linkages, in association with a guide-vane movement, causes the cam (15) to rotate, thereby initiating a signal from the position transmitter (16). This signal is fed into an error-detecting device (14) which, as a result of the imbalance introduced into the system, transmits a signal to the electro-hydraulic control unit (10), thus causing a displacement of this unit from the neutral position. This displacement is transmitted by a lever to the pilot valve (9), as a result of which, depending on the direction of motion, oil is either admitted to or exhausted from the upper side of the piston-type valve (2). This latter valve is in fact the main runner-blade regulating valve and any movement thereof relative to the central rod (3) allows oil to flow through ports A and B to and from the runner-blade servo-motor via the concentric control pipes located in the main-shaft bore. The resultant movement of the runner-blade servomotor piston is transferred back to the regulating valve by the control piping, which is rigidly connected to the piston at its lower end and to the restoring rod (3) at its upper end. Thus any movement of control valve (2) is immediately copied by restoring rod (3), the tendency always being to reclose ports A and B, thereby arresting the blade motion. Furthermore, any displacement of control valve (2) is conveyed to the position transmitter (8) by the feedback lever (7), and this gives rise to a second signal input to the error detector (14). The error detector compares and rectifies the signals it is simultaneously receiving from transmitters (8) and (16), and motion of the regulating valve (2) will continue until such time as these signals are in equilibrium.

It will be appreciated that the rate of movement of the runner blades depends on the rate of flow of oil to and from the regulating valve (2) via the pilot valve (9), and therefore the maximum opening and closing speeds of the blades can be adjusted by means of the throttle valves (11) and (12) respectively.

When valve (13) has been closed to isolate the electro-hydraulic control unit (10) from the circuit, the blade movement can be controlled by using the handwheel (5).

It should be noted that the central restoring rod (3) rotates with the

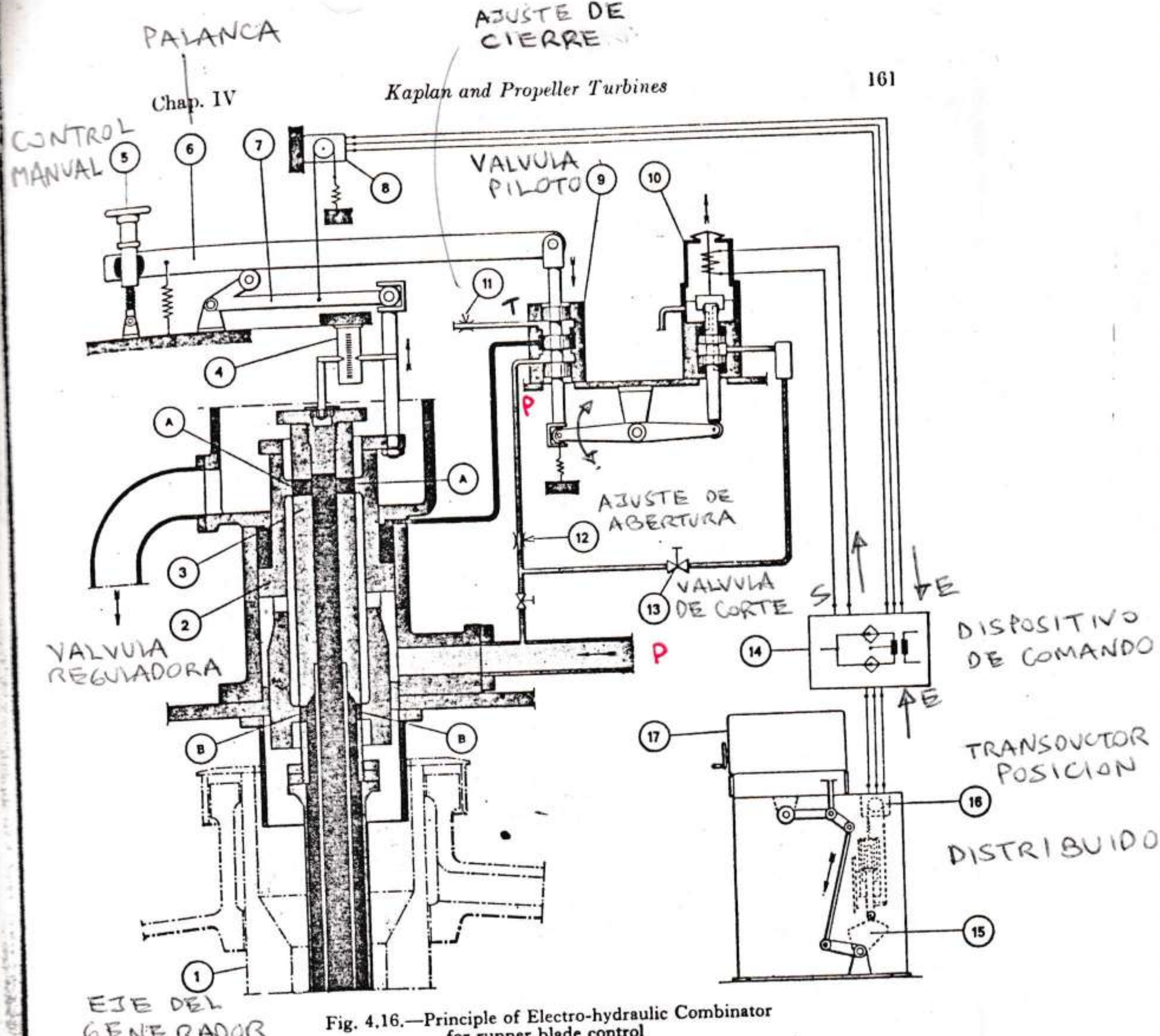

Fig. 4.16.—Principle of Electro-hydraulic Combinator for runner blade control

1. Generator shaft
2. Regulating valve
3. Restoring motion rod
4. Position indicator
5. Hand control device
6. Balance lever
7. Restoring motion lever
8. Position transmitter-restoring motion
9. Pilot valve
10. Electro-hydraulic control unit
11. Adjusting screw, closing speed
12. Adjusting screw, opening speed
13. Shut-off valve, control unit
14. Error detecting device
15. Combinator cam
16. Position transmitter
17. Actuator

A.B. Valve ports

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runner, being connected to the blade servomotor piston. The pointer of the blade position indicator (4) is located in the rod by means of a ball bearing.

The maintenance of the correct blade/guide-vane opening relationship is entirely dependent on the shape of the cam (15) and, where optimum efficiency is required over a wide range of heads, a three-dimensional cam would be installed with provision for lateral adjustment of the cam to produce different cam forms for each prevailing head.

One advantage of the design shown lies in the fact that the main regulating valve is easily accessible at the top of the generator shaft, whereas in older versions it was often the practice to locate the valve inside the runner hub.

For less-important Kaplan units the extra expense of the electrical-error detecting and transmission equipment may not be considered justified. In this case the system shown in fig. 4.16 can easily be converted to mechanical operation by omitting items (8), (10), (13), (14), (15) and (16) and replacing them by a connecting rod from the actuator linkage to a cam, located at the end of lever (6) remote from the pilot valve (9). This rod and cam now serve as the mechanical input signalling device to the pilot valve (9). With this sole exception the principle of operation is identical to that of the electro-hydraulic control mechanism.

In some power-station arrangements it may be considered undesirable to connect large and possibly unsightly pressure-supply and exhaust pipes from the station floor up to the top of the generator shaft. Furthermore, it may be found inconvenient in some designs of electrical machines to make provision for oil ducts within the generator-shaft bore.

These considerations, amongst others, have led to the relatively recent development of an alternative arrangement wherein the combinator system, comprising regulating valve, pilot valve, cam and linkages are all contained in a cabinet adjacent to the turbine pit. This cabinet may also contain the guide-vane servomotor regulating valve and associated pilot valve and linkages and, where applicable, the electro-hydraulic control unit. Thus a compact, centralized, hydraulic control cabinet is achieved, all parts of which are easily accessible for maintenance.

In such an arrangement the oil flow between the regulating valve and the runner-blade servomotor is effected through pipes connected

to a specially designed oil-head usually mounted on the turbine shaft immediately above the turbine guide bearing (fig. 4.1). In this case only the central bore of the turbine shaft is used for the oil ducts to and from the runner servomotor. The feedback linkage from the runner servomotor piston to the auxiliary pilot valve can also be designed into this oil head or, alternatively, can be taken from the guide-vane regulating ring.

4. Guide-vane Apparatus.

4.1. *Guide Vanes, Levers, and Links.*

The guide vanes as used for inside regulation (fig. 4.2) are normally of cast iron. No levers are used, a link being taken from a pin in the guide vane to the regulating ring. Bronze bushes are forced into each end of the guide vane to provide a bearing surface between the guide vane and the fixed guide-vane bolt.

Outside regulation requires a different design of guide vane, which is either of cast or fabricated steel construction. Rubber sealing strips are sometimes provided along the contact surface of the guide-vane blade in order to reduce water leakage to a minimum when the turbine is shut down.

The top and bottom guide-vane spindles turn in greased bronze-bushed bearings in the top and bottom covers. Recently new self-lubricating bearing materials such as P.T.F.E. or graphite-filled bronze have become available, and these are being used successfully for guide-vane spindle bearings, thus eliminating the need for grease connections. Teflon

The seals provided immediately below the lower bronze bushes must prevent all except the slightest leakage and should be of inverted U- or L-section and exert a slight outward pressure. When they are under water pressure, a good seal is thus obtained which for most designs will be drop-tight and cause the minimum of friction.

The guide-vane lever is usually of cast iron and keyed to the upper end of the spindle. At its other end an eccentric pin is provided so that a slight adjustment of each guide vane in its closed position is possible. The connecting link between each lever and pin on the regulating ring is made of cast iron and is designed to break readily if some object is caught between the guide vanes and prevents their closure. The links are designed so that replacement can be effected easily and quickly. Alternatively shear pins are used in some designs.

4.2. *Regulating Ring.*

The arrangement shown in most of the figures incorporates a roller race, on the centre line of which the connection is taken from the regulating ring to the servomotor by means of a rod. With inside regulation a similar arrangement can be used, the regulating ring being again supported on rollers as is shown in fig. 4.2.

In the plant shown in fig. 4.4 the regulating-ring support is bolted and spigoted on to the top cover, and it carries in turn the roller race and a regulating support-ring cover. These two parts are, with this design, made of cast iron with accurately machined surfaces where required. Steel rollers of two slightly different sizes are placed alternately in the annular space provided. Cages for one size of roller only are used for the larger turbines. The regulating ring, of cast iron or cast steel according to the loads to be carried, has pins bolted to the lower flange to which the guide-vane links are connected. At a suitable point the double flange opposite the roller race is widened and shaped to take the connecting-rod pin.

This method of construction has the advantage that only a single servomotor need be used. This simplifies the power-station structure, oil piping, and linkage to the actuator. Other designs which do not incorporate rollers are more commonly equipped with two servomotors.

5. Stay Ring and Spiral Casing.

For lower heads and large units, separate stays are provided which have their ends embedded in the concrete in the early stages of the power station construction (figs. 4.3 and 4.4). Their construction is either of solid bar, usually with welded-plate steel fairings, of cast iron or, where higher loadings are present, of fabricated or cast steel. Pads are fitted to the lower ends and these are drilled to take foundation bolts, the upper ends being arranged (fig. 4.3) either to be bolted to the wall frame or fitted with pads drilled for foundation bolts as in fig. 4.4. In section they are shaped so as to direct the water to the guide vanes with minimum loss. The number required for any unit is related to the number of guide vanes used; i.e. for 24 guide vanes 8 or 12 would be used according to the hydraulic and strength requirements.

With the independent stay arrangement, lip plates are required, partly to avoid difficult concrete shuttering but mainly to reduce water leakage and to present a smooth surface to the flow of water. Good-quality concrete work at the upper ends of the stays is essential to

prevent water leaking along the junction between the concrete, wall frame, and lip plate.

In the smaller sizes of turbine and for higher heads also, where inside regulation cannot be used or where outside regulation is specified, the stays are welded to lip plates of heavy section to form a stay ring. The stay ring is often in two or more sections according to transport requirements. Fig. 4.5 shows a typical arrangement of stay ring for a small turbine with a concrete spiral casing for operation under a medium head.

Where a plate-steel spiral casing must be used, the construction of the stay ring to which the steel casing is riveted or welded is rather different. Spiral casings are generally designed to withstand one and one-half times the maximum working pressure, and accordingly the stay ring must be designed to take the water-pressure loads from the platework, whatever the loads from the power-station superstructure may be.

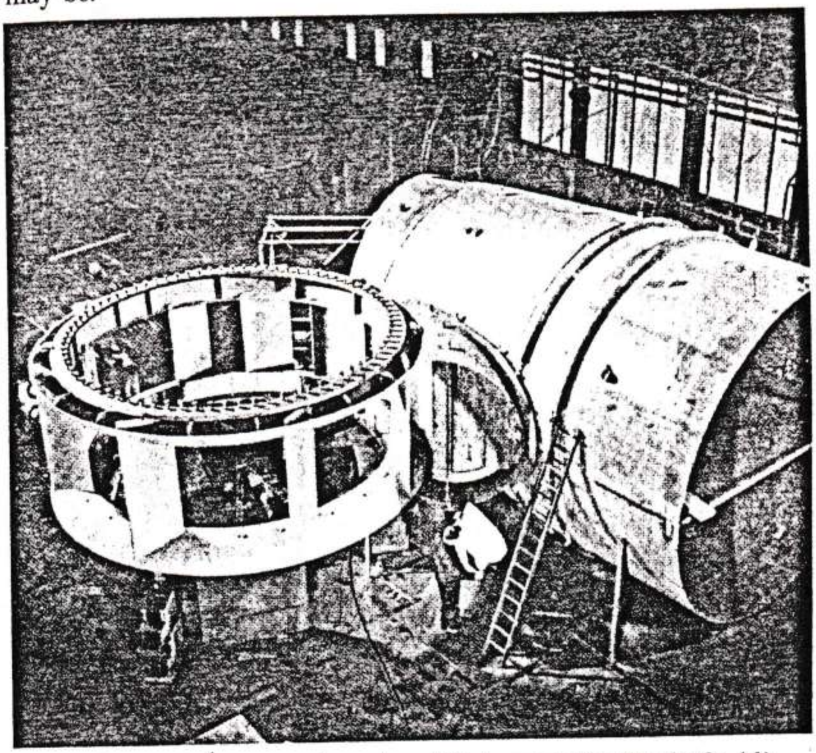

Fig. 4.17.—Spiral casing and stay ring of Kindaruma turbine (see also fig. 4.1)

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The stay ring is either of cast steel or fabricated design, and this choice should be left to the turbine manufacturer. It would be made in one or more pieces according to transport requirements. Figs. 4.1, 4.6, 4.17 and 4.18 show this type of construction.

In some installations the stay ring, or wall frame to which the stays are bolted, is arranged with pads to take a fabricated-steel structure which transmits the generator weight and part of the substructure load to the foundations (fig. 4.3).

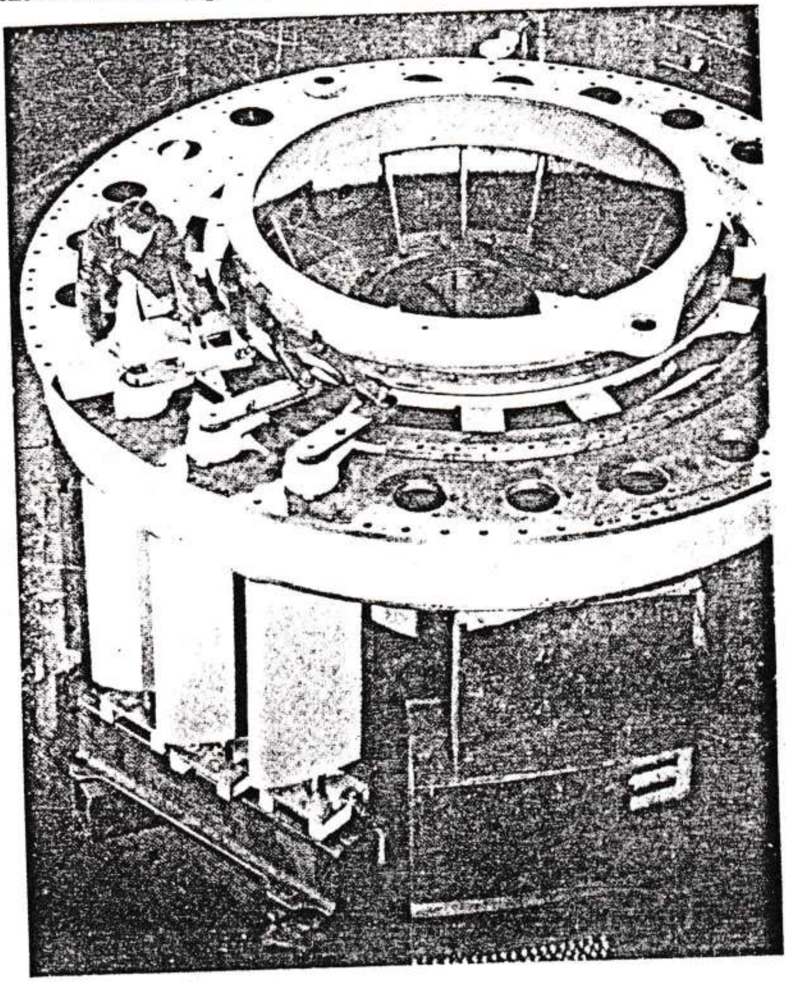

Fig. 4.18.—Top cover, regulating ring and guide apparatus of Kindaruma turbine (see also fig. 4.1)

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A design with alternate stay vanes pivoted has been developed for turbines for a canal scheme. The object of this arrangement is to close off the water from the guide vanes without shutting off water from the spiral casing, in order to enable the flow to be discharged from the spiral casing through a by-pass valve. Provided that only moderate leakage occurs the turbines can be given routine inspection without the necessity of by-pass gates, thus achieving a reduction in civil-engineering costs. This design calls for by-pass valves of completely reliable design, requiring only very occasional maintenance, since the turbines cannot be used during inspection or repair of the valves.

Since the water velocities through the stays are quite low, the finish of the stays or stay ring can be left as cast, after surface irregularities have been cleaned off.

6. Turbine Covers.

6.1. *Top Cover.*

The design of the top cover depends mainly on the turbine size, head, erection procedure, and partly on the method adopted for dismantling the runner blades.

In the smallest sizes the top cover is made in two concentric parts, each of which may be further split radially into two for ease of transport or erection. In fig. 4.2 an outer top (guide-vane) cover is provided to which the guide-vane bolts are attached. The inner part of the top cover, either in one or two pieces, rests on the outer top (guide-vane) cover. These two concentric cover parts must be dimensioned so that, when the inner top cover is removed, the complete runner can be lifted out without disturbing the outer top cover and guide vanes.

In medium and large-size units it is not essential to facilitate runner removal in this way, because their dimensions permit access to the runner blades and hub for inspection and maintenance, usually via removable sections of the runner chamber. Therefore we can find many examples of medium to large-size machines where the guide-vane cover and inner top cover have been combined in a single piece, usually split radially to ease dismantling (fig. 4.4).

On the other hand many purchasers prefer to pay the extra cost of the concentrically divided covers in order to preserve the facility of removing the runner for major repairs or overhaul without disturbing the guide apparatus (figs. 4.1 and 4.6).

For some very large turbines where handling of the heavy blades within the confines of the runner chamber becomes impracticable, the

concentrically-split top cover again becomes an unavoidable additional expense.

The top cover carries iron bushes with upper and lower bearing liners for the upper end of the guide-vane spindles and is designed to withstand, with only small deflection, the water pressure and other loads. For smaller units and over the lower range of heads it is generally of cast iron, but a fabricated steel-plate construction is more often used for medium to large-size units with higher operating pressures. The cover is provided with integral machined flanges which mate with the guide-bearing support brackets and the regulating ring support (fig. 4.4).

On most machines a separate inner cover is located immediately above the hub, bolted to the underside of the top cover and hydraulically shaped to lead the water smoothly on to the hub and blades. This cover is often equipped with hand holes with removable covers so that access can be gained to the shaft seal by men working from the dewatered runner space (fig. 4.6).

The wetted surface of the top cover need be machined only over that part adjacent to the guide vanes. A ridge can be formed along the pitch-circle diameter of the guide vanes so that the clearance through which leakage water can flow, with closed guide vanes, is small. From this area to the hub the surface finish need be only as cast with major surface irregularities removed or rough-machined.

6.2. *Bottom Cover.*

For small cheap units for low-head installations with inside regulation, the lower guide-vane cover and runner chamber are usually made as one part in cast iron (fig. 4.2).

However, for heads above 6 m. and for any but the smallest machines, the runner chamber is invariably separated from the cast-iron bottom cover and made of a material more resistant to pitting.

The wetted surface of the bottom cover should be smooth-machined.

7. Runner Chamber.

The shape of the runner chamber, for all except the lowest heads, is usually cylindrical above and spherical below the runner centre-line, although practice varies with different turbine manufacturers. For very low heads a cylindrical runner-chamber is generally adopted, since the clearance losses will be small in proportion to the total losses, and this shape facilitates machining. Although cast iron is satisfactory for the lowest heads it should not be used where there is any possibility of

pitting, since satisfactory repairs cannot be effected readily. A cast or fabricated steel design can be used for heads up to approximately 18 m. beyond which cast stainless steel or stainless-steel-clad plate has been found to be more satisfactory. Where the water carries silt it is advisable to use stainless steel.

It is usual practice to make a section of the runner chamber removable to facilitate inspection of the runner and removal of the blades without dismantling the generator. With a completely spherical runner chamber the upper part must be removable, an operation for which it is difficult to provide in designs for small turbines.

For propeller turbines a cylindrical runner-chamber is generally used although an expanding shape has been adopted in a few cases. An accurately machined shape with a smooth ground finish is always required for the runner chamber.

8. Draft Tube.

8.1. *General Design.*

The efficiency of the draft tube is especially important for Kaplan turbines, in particular at low heads, owing to the high water velocity at the runner throat. One of the major problems in the early development of Kaplan and propeller turbines was the design of an efficient draft tube to represent the best compromise between turbine performance and civil-engineering construction costs. Some turbine designers believed that division of the flow by horizontal and vertical walls was essential. While from the construction point of view such arrangements were of some advantage, the present trend is now generally confined to a vertical dividing wall over a short length of the horizontal or radial portion at the exit end.

It is very seldom possible to use the ideal straight conical type of draft tube in order to convert the kinetic energy to pressure head, owing to the great depth of excavation required. A right-angled bend must be located some distance below the runner and even this requires a considerable depth of excavation for units of large output at higher heads. The losses in this bend will depend on the velocity, and some compromise between turbine efficiency, excavation, and plant costs must be made. It is therefore important that the velocity is reduced, in as short a distance as possible, between the runner and the bend, particularly for high-specific-speed installations where draft-tube efficiency has a more marked effect on turbine efficiency. The effect of too short an axial distance between runner and bend is to reduce the

efficiency over the whole load range, particularly at full load, and also therefore to reduce the maximum output.

The radial length of the draft tube, i.e. the distance between the turbine axis and draft-tube exit, also has an appreciable effect on the efficiency curve. The effect on the peak efficiency is small but towards, and particularly at, maximum output the reduction in efficiency and output is quite marked as the radial length is reduced. The increase in area required to reduce the exit loss cannot be obtained in less than a certain distance because, with too great a rate of increase in area, eddy losses more than offset the gain due to reduced velocity head at the draft-tube exit.

It should be noted that for a conical section of draft tube with a total angle of divergence somewhat greater than the ideal for axial flow of approximately 7°, an improvement in efficiency is obtained by designing for a small swirl component. Research has shown that the best efficiency of a Kaplan or propeller turbine does not occur with axial flow from the runner, but with a small swirl component in the same direction as that of runner rotation. This has led to an improvement in turbine efficiency for a given depth of excavation.

8.2. *Throat Ring and Draft-tube Cone.*

Where a cylindrical runner chamber is used, no throat ring is required, the draft-tube cone being immediately adjacent.

For high heads a throat ring is provided, at which section the change in shape from the spherical runner chamber to the conical upper end of the draft tube takes place. For all heads up to approximately 18 m., it would be made of cast steel, i.e. of the same material as the runner chamber. As the possibility of pitting of its surface is only a little less than that of the runner chamber it is advisable to use the same material. The surface finish should be the same as that of the runner chamber.

It is usual practice to use either the throat ring or the draft-tube cone for turbines with cylindrical runner chambers as a starting point for erection. The initial concrete is placed and steel joists embedded in it so that accurate location of the throat ring is facilitated. For this purpose these parts are provided with brackets and adjusting screws and anchor irons, preferably of round bar but sometimes of plate strip. As for the runner chamber, it is best to use round anchor bar screwed into holes provided in bosses suitably spaced on the throat ring. Such a construction ensures a rigid location of parts subject to vibration if the safe cavitation factor is exceeded.

The draft-tube cone is generally of heavy mild-steel plate in its upper section and in two or more parts, except for the smallest size where transport and/or erection requirements permit its construction in one piece.

8.3. *Draft-tube Bend.*

The shape shown in fig. 4.23 is one of the original designs used for the Kaplan turbine and has the advantage that it simplifies the concrete shuttering. It is usually only necessary to provide steel protection for the concrete at the roof of the bend from the high water velocities. The upper part of the bend, which is curved in two directions, is normally supplied with the turbine. Where sand, gravel, or heavy silt in the water is liable to cause deterioration of the concrete surface, it is advisable to include a complete lining extending round the elbow to the end of the bend. Such an extension may have a secondary advantage of providing a formwork for the concrete thereby saving some of the more complicated shuttering. It is always worth investigating whether for this reason alone an extension of the draft-tube lining is justified. A reasonably smooth inside surface should be provided for the draft-tube bend lining, and welding is therefore the more suitable method of fabrication. As for the draft-tube cone, ample anchor iron should be provided.

For further information on draft-tube design the reader is referred to Chap. III, § 6.

9. Bearing and Lubrication.

9.1. *Sleeve-type Bearings.*

Turbine guide bearings may be lubricated either by grease or oil. Grease lubrication has been used in the past for sleeve-type bearings from the smallest to the largest sizes of shaft (fig. 4.7) but the disadvantage of continual wastage of grease and the need for a higher rate of heat dissipation from modern high-speed bearings have led to a general preference for the more expensive design using oil as the lubricant.

In the earlier forms of oil-lubricated sleeve bearings the continuous circulation of oil between the bearing lining and the shaft was achieved by means of motor or gear-driven pumps which pumped oil from a sump below the bearing to a reservoir above it, whence it flowed under gravity through the bearing itself back to the sump (fig. 4.5). For larger machines it was common practice to ensure uninterrupted oil circulation

by using two pumps, one motor-driven and the other gear-driven from the shaft itself.

Another method once commonly used for the smaller machines was to attach the lower oil sump to the shaft so that it rotated with the shaft. A Pitot tube dipping into the rotating oil then utilized the dynamic head available to lift the oil to the top reservoir.

A modern development of this principle is the self-lubricated bearing which avoids the use of any separate pumps and which has proved itself so reliable as to be universally applied even to the largest machines (see figs. 4.1, 4.4 and 4.6). In this design the journal of the bearing takes the form of an inverted L-shaped collar forged integrally with the turbine shaft. The inner wall of the lower oil sump is inserted upwards into the annular space between the vertical leg of the collar and the shaft, so that the lower part of the journal is always immersed in oil. The mating surface of the bearing sleeve is provided with a number of grooves running from bottom to top and inclined in the direction of shaft rotation. When the shaft rotates, the journal surface drags the oil into these grooves, and the dynamic head of the rotating oil is sufficient to create a pumping action causing oil to flow through the bearing to the upper oil reservoir. The pumping action is so intense in well-designed grooves that adequate oil circulation is initiated even at very low rotational speeds, thus eliminating any danger of seizure of the bearing when starting the machine. The only safety devices required are a low-oil-level alarm in the oil sump to give warning of insufficient submergence of the lower end of the journal, together with the usual temperature-recording instruments. The bearing is usually water-cooled by coils located either in the lower sump or in the upper reservoir.

9.2. *Pad-type Bearings.*

In spite of the relative simplicity and efficiency of the modern self-lubricated sleeve bearing, there is an increasing trend at the present day towards the use of the pivoted-pad bearing commonly known in this country as the Michell bearing. In this respect turbine engineers have lagged behind their associated generator manufacturers who adopted the pad bearing at a much earlier stage.

The principle of this type of bearing is well known, and it is sufficient merely to draw attention here to the fact that no upper oil reservoir is necessary, oil circulation between pads and journal being maintained simply by the rotation of the shaft itself. To avoid the use of expensive

and unreliable oil seals between the bearing casing and the shaft, it is common practice to provide the same inverted L-shaped journal described earlier and to design the inner wall of the bearing sump sufficiently tall so that the surface of the oil-bath can never overtop it.

The main advantage of the pad-type bearing lies in the fact that by means of the individual adjusting screws provided on each pad pivot, the bearing clearances can be varied at will to suit operational experience. Furthermore, removal and replacement of the pads is extremely simple and inexpensive compared to the maintenance and repair of the grooved sleeve of the self-lubricated sleeve bearing.

10. Shaft Gland.

Since the runner centre-line of a propeller or Kaplan turbine is usually below tailwater level, it is necessary to have a positive shaft seal to prevent the entry of water into the bearing and inner cover. The carbon-ring seal is frequently used and requires little attention for long periods. The wear on the carbon blocks is automatically taken up by garter springs which maintain the pressure of the blocks on a renewable sleeve which is made in halves and bolted to the shaft. It is sometimes considered beneficial to provide a labyrinth just below the seal to prevent the possible entry of dirt which would lead to undue wear of the sealing surface. Where appreciable quantities of suspended matter are present a supply of clean water is fed to points between the labyrinth and first sealing ring.

11. Intake Arrangements.

The design of the intake for Kaplan and propeller turbines is, within limits, mainly a matter of economics. The maximum water velocity in the intake is determined by consideration of head losses and initial cost. In general this velocity varies between 0·05 and $0{\cdot}2\sqrt{(2gH)}$ at rated output and head.

Some typical intake arrangements are shown in fig. 4.24. For the highest heads a plate-steel spiral is generally used as in (*a*). Where good-quality concrete work is certain and the amount of reinforcement is moderate, a rectangular concrete spiral casing is used, sometimes even for heads as high as 35 m. as in (*b*). Below heads of 20 m. large units are usually arranged with an intake similar to that shown in (*c*). In schemes where the turbine dimensions and output are small an open setting or simplified concrete casing can be used as in (*d*).

12. Synchronous-condenser Operation.

If the generator is required to operate sometimes as a synchronous condenser, the Kaplan turbine has an advantage in that the blades, when fully closed, do not give rise to as large a windage loss as occurs with a propeller or Francis turbine under the same operating conditions. On the other hand, except with very low-head installations, the runner will be placed below the tailwater level. This condition generally necessitates some means of admitting compressed air to the runner chamber to depress the level of the water in the draft tube to enable the blades to rotate in air. In resuming normal generation the guide vanes should not be opened too quickly owing to the possibility of causing abnormal vibration due to rapid removal of the air by the water entering the runner chamber.

In order that the losses shall be reduced to a minimum, it is essential that the blades are in the fully closed position, for which upward thrust due to the runner rotating in water behind closed guide vanes is also a minimum. This requirement makes it necessary to provide equipment to ensure that any automatic blade-resetting mechanism used for starting up is inoperative when the guide vanes are fully closed and the runner is rotating.

An effective shaft seal must be provided to prevent excessive air leakage, and it is also desirable to have sealing inserts in the guide vanes to reduce leakage of water to a minimum. It would appear that the leakage water due to full water pressure in the spiral casing or concrete intake chamber has the effect of an ejector. The air leakage, however, through a well-designed shaft seal is comparatively small.

The equipment used for providing the compressed air for lowering the water level in the runner chamber varies considerably. In some plants several air vessels are used in conjunction with high-pressure compressors. In others a high- and low-pressure system is used; high pressure for supplying a large quantity of air quickly for the rapid lowering of the water level, and the low-pressure system for making good the leakage. Some plants are supplied, however, with low-pressure compressors to provide air for lowering the water level slowly and to make up for leakage.

Some authorities, while not requiring synchronous-condenser operation, prefer to install the equipment required to enable a machine to motor on the system with the turbine guide vanes closed. This enables the set to pick up load comparatively quickly without the delay of synchronizing.

13. Selection of Speed, Runner Level and Turbine Dimensions.

13.1. *Choice of Speed and Setting.*

The decision to install Kaplan turbines in any scheme must be made with a full appreciation of their hydraulic characteristics, and a clear indication of the various requirements must be given to the designer, since sometimes certain of them conflict. For example, the requirements that a turbine shall operate frequently at low loads and yet have only shallow excavation for the draft tube are in conflict; a high-specific-speed turbine, with dimensions smaller than normal and therefore with a high cavitation factor, would meet the first requirement but it would not satisfy the second because a deep setting is required to give the requisite suction head.

The requirement that best efficiency should be achieved close to or at full load with a turbine of normal specific speed, will lead to the use of a design of runner best suited to a higher head and one which will also be appreciably larger than for a normal design.

If full output is required over a range of head varying by more than about five per cent from the design head, the turbine dimensions will increase as the head variation increases. Further, to obtain good efficiencies over the head range, a peripheral velocity lower than normal will be required and the specific speed will therefore be lower. Where, however, the headwater level is almost constant and variations in net head of not more than 10 per cent are created by fluctuations of the tailwater level such as occur during floods, full output can generally be obtained without departure from the normal design, other than a small increase in the maximum runaway speed.

When the headwater level varies appreciably it is more usual to limit the flow through the turbine to a figure corresponding to a lower head, in which case the cavitation limit at the lower operating heads will generally determine the runner setting. Such a requirement can usually be met with a moderate decrease in the specific speed at the design head.

The information given below is intended as a reasonably accurate guide to estimating the main dimensions and setting of Kaplan turbines for normal schemes. The difference between preliminary estimates and the final design will be small in most cases but may be appreciable where unusual requirements must be met.

Fig. 4.19 (for Kaplan turbines) and fig. 4.20 (for propeller turbines) give curves of both specific speed and a constant

$$C = (\text{runner diameter}/\sqrt{\text{horse-power}})$$

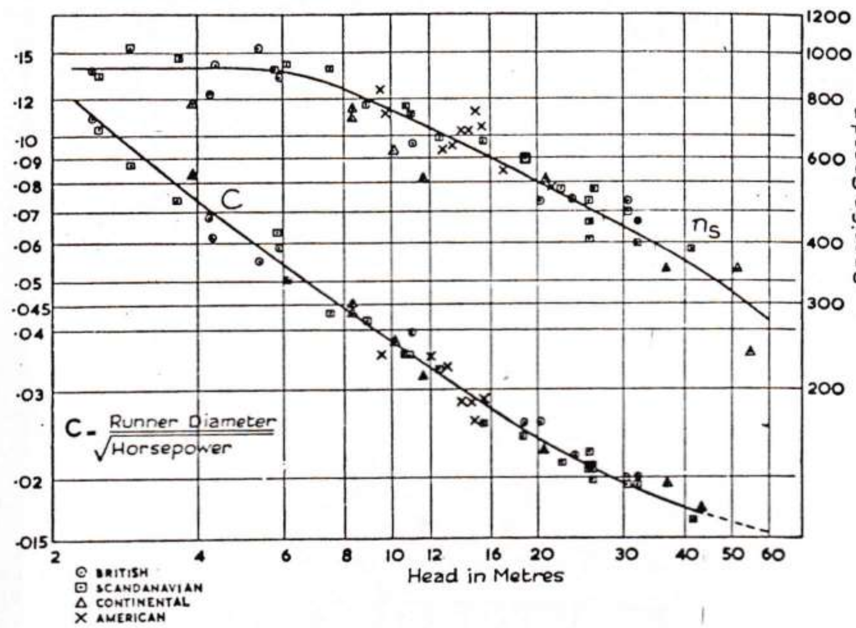

Fig. 4.19.—Curves of specific speed, head and runner diameter (Kaplan turbines)

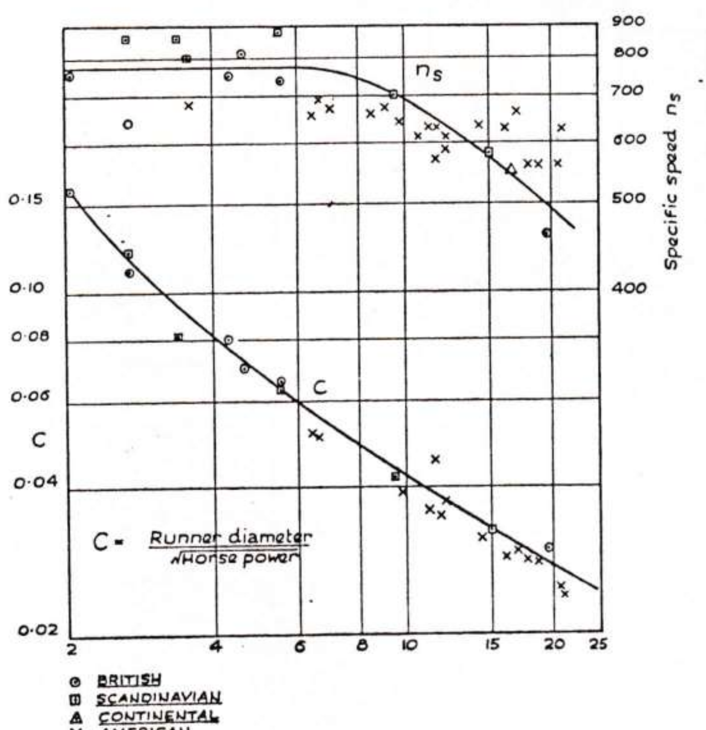

Fig. 4.20.—Curves of specific speed, head and runner diameter (propeller turbines)

against head for a number of existing turbines. It will be seen that while the specific speed varies considerably between different designs, the variation of runner diameter is much smaller. The estimation of the latter is therefore less liable to inaccuracies than the specific speed. The head for which full-load output is required is used for reading off the constant. It is assumed that this head also corresponds to that at which the best efficiency is required. If these two heads are not the same an adjustment to the speed as found from the upper curve must be made. For example, if full output is required at 25 m. but best efficiency at 28 m. then the speed is increased in the ratio of the square root of the heads. A check should be made to see that the specific speed at the lower head does not exceed the limits of normal practice by more than about 5 per cent.

The runner setting or suction head is generally expressed as the distance between the runner centre-line and minimum tailwater level. In most cases (low-head schemes being a general exception) this distance is a minus quantity and the runner is below tailwater level. The minimum tailwater level is usually taken as that corresponding to one

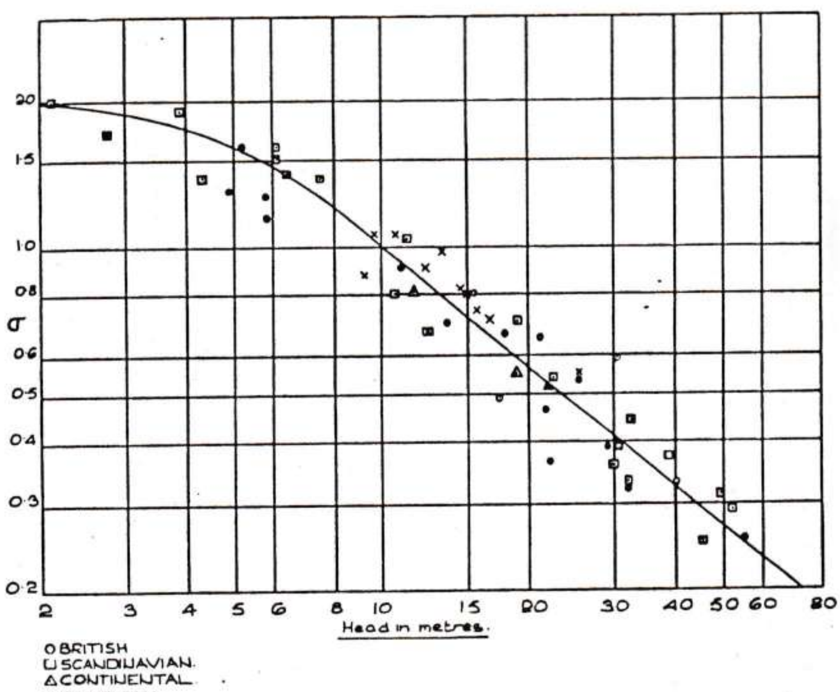

Fig. 4.21.—Cavitation factor and operating head for Kaplan turbines (for propeller turbines reduce by 8%)

unit operating at rated head and output. Where the river bed is likely to be eroded, an estimate of the future tailwater rating curve should be made in order that the suction head is not exceeded later during the operating life of the plant.

Fig. 4.21 gives a curve of cavitation factor plotted against head, with a number of points corresponding to existing plants. Turbines operating under normal conditions would require an average value.

The suction head is then estimated as follows:

$$H_s = H_B - (\sigma H + H_v + H_1)$$

where σ = cavitation factor,
H_B = barometric pressure at runner elevation, m.,
H = net head, m.,
H_v = vapour pressure of water, m.
H_1 = height of runner-blade leading edge above runner centre-line, m.,
= $0{\cdot}15D$ m. (for vertical-shaft units),
= $0{\cdot}5D$ m. (for horizontal-shaft units).

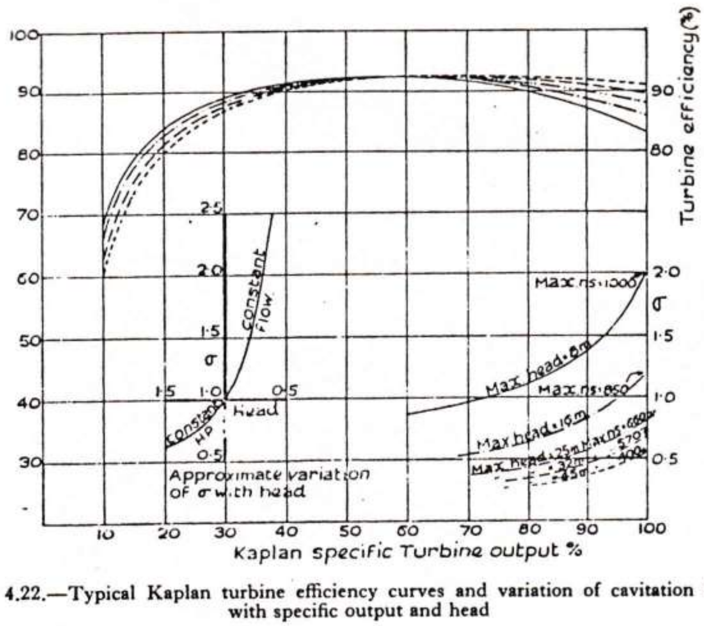

Fig. 4.22.—Typical Kaplan turbine efficiency curves and variation of cavitation factor with specific output and head

Typical efficiency curves are given in fig. 4.22 with corresponding sigma values for a Kaplan turbine giving an output of approximately

10,000 h.p. The efficiency curves for various operating heads show reduced full-load efficiency as the head is lowered. Best efficiency occurs at about 0·6 and 0·8 of full-load output for low- and high-head schemes respectively.

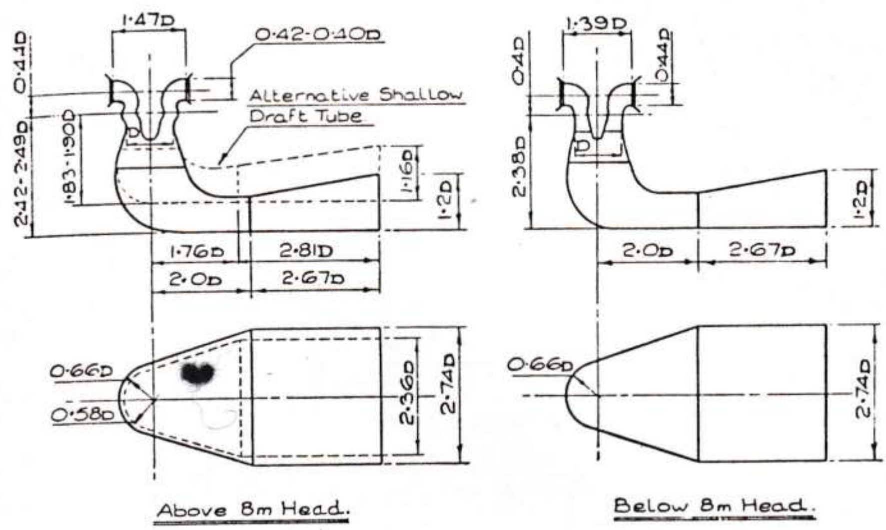

Fig. 4.23.—Turbine and draft-tube dimensions

13.2. *Turbine and Intake Dimensions.*

An approximate outline of the power-station substructure can be prepared from the estimated turbine size and setting together with that of the generator size. Space must, of course, be allowed for ancillary equipment. Fig. 4.23 gives the main dimensions of the turbine and draft tube, and fig. 4.24 gives the main dimension of the intake.

Where a shallow excavation for the draft tube is required it must be noted that the loss at best efficiency varies from 0·5 per cent at 30 m. head to 1·0 per cent for the lowest heads. For higher heads the difference is progressively smaller. At full load the loss is about twice these values.

13.3. *Example* 1.

A Kaplan turbine is required, output 15,600 h.p., net head 25 m. Head variation can be neglected. Excavation must be kept to a minimum. Tailwater level at full load = 150 m. above mean sea-level. Water temperature = 20° C.

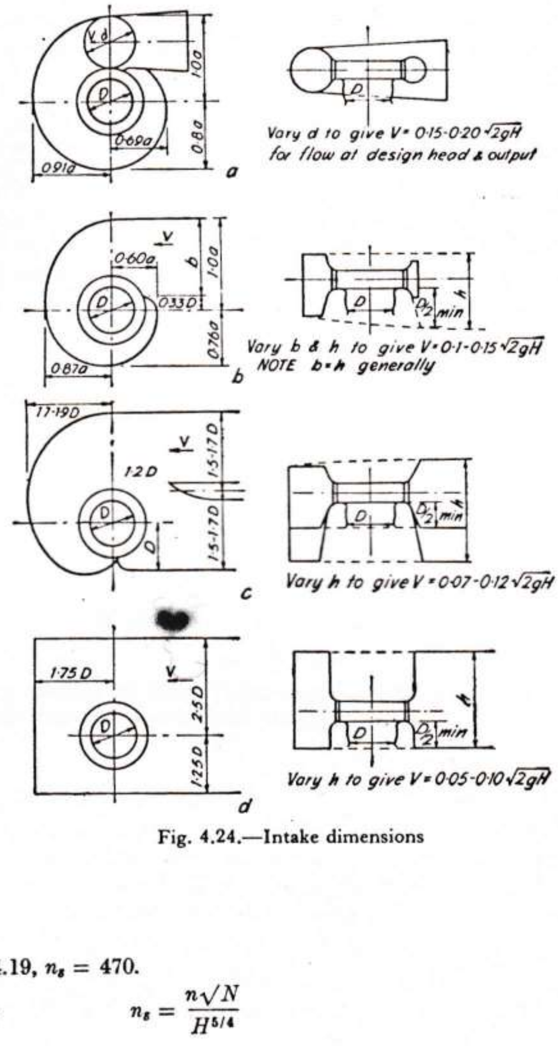

Fig. 4.24.—Intake dimensions

Speed.

From fig. 4.19, $n_s = 470$.

Using formula $$n_s = \frac{n\sqrt{N}}{H^{5/4}}$$ (see p. 5)

$$n = \frac{470 \times 25^{5/4}}{\sqrt{15{,}600}} = 211 \text{ r.p.m.}$$

Nearest synchronous speed = 214 r.p.m.

Runner diameter.

From fig. 4.19, $C = 0{\cdot}022$.

Using formula, runner diameter $= C\sqrt{N}$

$= 0{\cdot}022\sqrt{15{,}600}$

$= 2{\cdot}75$ m.

Runner setting.

From fig. 4.21, $\sigma = 0{\cdot}49$ and from fig. 4.25, $H_B = 10{\cdot}1$ m. $H_v = 0{\cdot}22$ m. Using formula, suction head $H_s = H_B - (\sigma H + H_v + H_1)$

$$= 10{\cdot}1 - (0{\cdot}49 \times 25 + 0{\cdot}22 + 0{\cdot}41)$$
$$= -\ 2{\cdot}78 \text{ m.}$$

Elevation of runner centre-line $= 150 - 2{\cdot}78 = 147{\cdot}22$ m.

Spiral casing dimensions.

From fig. 4.22 full-load efficiency will be approximately 91 per cent with a normal draft tube. As, however, a shallow draft tube is required, the efficiency will be reduced to about 90 per cent. At full load the turbine will discharge $53{\cdot}2$ m^3/s. Choosing a spiral casing velocity of $0{\cdot}18\sqrt{(2gH)}$ (see fig. 4.24a)

$$d^2 = \frac{53{\cdot}2}{0{\cdot}18\sqrt{(19{\cdot}62 \times 25)}} \times \frac{4}{\pi}$$
$$d = 4{\cdot}1 \text{ m.}$$

Referring to gs. 4.23 and 4.24 (a)

$$a = 4{\cdot}1 + \tfrac{1}{2}(1{\cdot}47 \times 2{\cdot}75) = 6{\cdot}12 \text{ m.}$$

Draft-tube dimensions.

From fig. 4.23, distance between runner centre-line and floor of draft tube is

$$1{\cdot}90 \times 2{\cdot}75 = 5{\cdot}22 \text{ m.}$$

Elevation of draft-tube floor $= 147{\cdot}22 - 5{\cdot}22$ m.

$$= 142{\cdot}0 \text{ m.}$$

Note.—For heads above 18 m. constants 0·40, 1·19 and 2·49 should be used in estimating the turbine and draft-tube dimensions.

Distance between units.

If two or more units are required, the distance between units may be estimated from fig. 4.24 with an allowance for minimum space between the spiral casings for thickness of concrete. This distance, which may have to be increased to allow for the location of ancillary equipment, will in this case be

$$1{\cdot}80a + 0{\cdot}8 = 1{\cdot}80 \times 6{\cdot}12 + 0{\cdot}8 = 11{\cdot}8 \text{ m.}$$

the figure of 0·8 m. being an allowance for the minimum distance which will permit welding of the spiral casing at site.

13.4. *Example* 2.

A Kaplan turbine, output 3500 h.p., is required to operate with a net head varying between 26 and 6 m. Best efficiency is required at 20 m. Between 20 and 12 m. the water quantity is to be constant corresponding to an output of 3500 h.p. at 20 m. Between 12 and 6 m. the output and water quantity is to be limited to that which can be obtained without cavitation. Tailwater level is constant at 300 m. above mean sea-level. Water temperature = 18° C.

Runner diameter $= 0{\cdot}0245\sqrt{3500} = 1{\cdot}45$ m.
Specific speed $= 533$

$$\text{Speed} = \frac{533 \times 20^{5/4}}{\sqrt{3500}} = 381$$

Nearest synchronous speed = 375 r.p.m.

Runner setting.

In order to determine the safe suction head a check must be made over the operating conditions specified to see that the correct sigma is chosen.

At 20 m., $\sigma = 0{\cdot}56$

$$H_1 = 0{\cdot}15 \times 1{\cdot}45 \text{ m.} = 0{\cdot}22 \text{ m. } H_v = 0{\cdot}20 \text{ m. } H_B = 10{\cdot}0 \text{ m.}$$

$$\therefore H_s = 10{\cdot}0 - (0{\cdot}56 \times 20 + 0{\cdot}22 + 0{\cdot}20) = -1{\cdot}62 \text{ m.}$$

At 12 m. From fig. 4.22,

$$\text{Head} = \frac{12}{20} = 0{\cdot}60.$$

$\therefore$ Correction to sigma $= 2{\cdot}5$,
i.e. new sigma value $= 2{\cdot}5 \times 0{\cdot}56 = 1{\cdot}40$.
$\therefore H_s = 10{\cdot}0 - (1{\cdot}40 \times 12 + 0{\cdot}22 + 0{\cdot}20) = -7{\cdot}22$ m.

At 26 m. Head $= \frac{26}{20} = 1{\cdot}30$.

$\therefore$ Correction to sigma $= 0{\cdot}75$,
i.e. new sigma value $= 0{\cdot}75 \times 0{\cdot}56 = 0{\cdot}42$.
$\therefore H_s = 10{\cdot}0 - (0{\cdot}42 \times 26 + 0{\cdot}22 + 0{\cdot}20) = -1{\cdot}34$ m.

The suction head as found for 12 m. head is clearly too high and would involve expensive excavation. The specific speed would have to be reduced to obtain a reasonable suction head. Assuming that a value of 3 m. would give the lowest total cost for the scheme considered, the new speed and runner diameter can be estimated as follows:

$$H_s = -3 = 10 - (12 \times \sigma + 0{\cdot}22 + 0{\cdot}20)$$

$$\therefore \sigma = \frac{12{\cdot}68}{12} = 1{\cdot}06$$

Dividing this by the factor 2·5 to arrive at the sigma for full-load output at the design head we obtain

$$\sigma = \frac{1{\cdot}06}{2{\cdot}5} = 0{\cdot}425$$

Referring to fig. 4.21, this value of sigma corresponds to a head of 28 m. which gives the new specific speed

$$n_s = 450.$$

$$\text{Speed} = \frac{450 \times 20^{5/4}}{\sqrt{3500}} = 321.$$

$\therefore$ Synchronous speed = 333 r.p.m.

As the specific output decreases with increased head, and as the output is proportional to the square of the diameter, a correction must be made to the diameter first estimated, i.e. 1·45 m.

From fig. 4.19 at 28 m., $C = 0{\cdot}020$
and at 20 m., $C = 0{\cdot}0245$.

$$\text{Correction} = \frac{0{\cdot}0245}{0{\cdot}020} = 1{\cdot}225$$

i.e. new diameter $= 1{\cdot}45 \times 1{\cdot}225 = 1{\cdot}78$ m.

Pitch-circle diameter of trailing edge of stay vanes.

From figs. 4.23 and 4.24

$$1{\cdot}47D = 2{\cdot}62 \text{ m.}$$

Intake dimensions.

Taking full-load efficiency of the turbine as 89 per cent at 20 m., the turbine discharge is 14·9 m^3/s. As the head range is great and as a reduction in output and efficiency results if the intake velocity is unduly high, a coefficient of 0·10 is chosen.

$$\text{Velocity} = 0{\cdot}10 \times \sqrt{(2g \times 20)} = 1{\cdot}98 \text{ m/s.}$$

$$\therefore \text{ Required area} = \frac{14{\cdot}9}{1{\cdot}98} = 7{\cdot}53 \text{ m}^2.$$

Take $b = h = 2{\cdot}75$ m.

$$\therefore\ a = 0{\cdot}33 \times 1{\cdot}78 + 2{\cdot}75 = 3{\cdot}34 \text{ m.}$$

For this head a concrete spiral could be used and the intake could be outlined and other dimensions and levels estimated as in Example 1.

13.5. *Example* 3.

Three propeller turbines are required to give an output of 24,000 h.p., each under a net head of 21 m. Space requirements are such that the distance between units is to be kept to the minimum, even if this results in a wider power station.

Speed.

From fig. 4.20 $n_s = 480$

$$n = \frac{480 \times 21^{5/4}}{\sqrt{24{,}000}} = 139.$$

Runner diameter.

From fig. 4.20 $C = 0{\cdot}026$.

$$\text{Diameter} = 0{\cdot}026 \times \sqrt{24{,}000} = 4{\cdot}0 \text{ m.}$$

Runner setting.

From fig. 4.21 $\sigma = 0{\cdot}54 - 8\%$.
$= 0{\cdot}50$.

The suction head is then calculated as in Example 1.

Intake dimensions.

For this head and size of turbine the choice of intake arrangement will be between those shown in fig. 4.24 *b* and *c*. As the distance between units is to be kept as small as possible, arrangement *c* is better, although the stresses in the concrete will be high, thus necessitating more than normal reinforcement.

Assuming a full-load efficiency of 87 per cent the turbine discharge is 97 m^3/s.

With a propeller turbine it is better to choose a rather lower water velocity in the intake than for a Kaplan turbine, which operates for a larger proportion of the time at a load lower than that for a propeller turbine. Taking a coefficient of 0·085 for this preliminary estimate of power-station substructure dimensions the required area becomes

$$\frac{97}{0{\cdot}085 \times \sqrt{(2gH)}} = 56 \text{ m}^2$$

$$\text{Intake width} = 3D = 3 \times 4{\cdot}0 = 12 \text{ m.}$$

$$\text{Intake height} = \frac{56}{12} = 4{\cdot}66 \text{ m.}$$

Allowing 2 m. minimum thickness of concrete, the distance between units =

$$12 + 2 \fallingdotseq 14 \text{ m.}$$

The thickness of the intake dividing wall should not be less than about 3 m. in this case, but will depend on the load to be carried from the power-station superstructure and on the type and arrangement of intake gates used.

14. Notes on Choice of Kaplan and Propeller Turbines.

The operating experience at numerous power stations equipped with Kaplan turbines indicates that no special difficulties exist in using this type of turbine up to a head of approximately 50 m. for the largest outputs. Units are in operation for outputs as large as 110,000 h.p. under a head of 49 m. and 150,000 h.p. under a head of 36·5 m., and there are some smaller units operating at heads up to 73 m.

Propeller turbines have found favour mainly in North America for heads up to a maximum of approximately 30 m. They have not, however, found such marked favour in British practice and have generally been used only for low-head developments with low outputs.

The present trend is towards the use of larger and fewer units of the Kaplan type, which is adopted for most low- and medium-head schemes except for the smallest outputs. Where its use is considered as an auxiliary or house set, it must be remembered that it cannot be controlled entirely by hand as is possible for a small Francis turbine, owing to the necessity of having a supply of oil under pressure for the control of the runner blades, unless a hand-operated pump is provided for initial starting-up.

For a large scheme using Kaplan turbines it is of importance that limitations of maximum size and weight are known, since this is the limiting factor of the output that can be obtained from one unit. The largest and heaviest single part is the hub casting which cannot be split into parts to facilitate transport. The present general limit upon hub diameter is between 3 and 3·4 m. giving theoretically units with maxi-

mum outputs of roughly 20,000 h.p. at 5 m., 85,000 h.p. at 20 m., and 125,000 h.p. at 30 m.

For low heads up to approximately 20 m., and where large water quantities are to be utilized, the advantages of Kaplan turbines are quite clear; but as the head increases the plant costs are greater relative to those associated with Francis turbines. This is primarily due to the proportionately larger hub required for an increased head which,

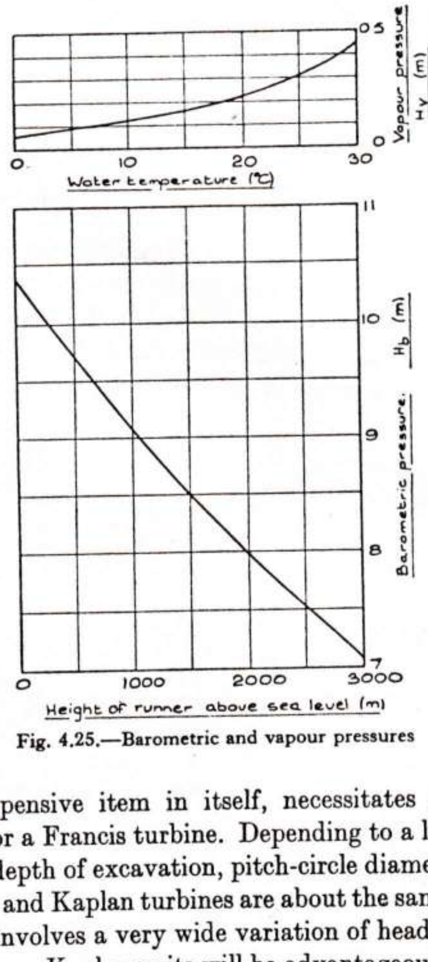

Fig. 4.25.—Barometric and vapour pressures

although an expensive item in itself, necessitates a larger runner diameter than for a Francis turbine. Depending to a limited extent on the permissible depth of excavation, pitch-circle diameters of the guide vanes of Francis and Kaplan turbines are about the same at 20 m. head.

If a scheme involves a very wide variation of head, say as much as two or three to one, Kaplan units will be advantageous, even for heads higher than 30 m., owing to the smaller change of peak efficiency over the head range compared with a Francis or propeller turbine. This

generally leads to the use of Kaplan units for variable-head schemes up to a present maximum of about 73 m.

Kaplan units are of particular advantage for run-of-river schemes where storage is small and the headwater level must be maintained constant within narrow limits. With low flows the units can operate at best efficiency, and under flood conditions, when the tailwater rises, the high overload capacity enables full output to be obtained without cavitation and without departure from the normal design for a reduction in head of as much as 10 per cent which may result from an increase in tailwater level.

15. Typical Representative Schemes.

15.1. *Vargön Power Station.*

This almost unique installation, shown in fig. 4.26, is an example of the economic development of a very large water quantity available at a low head. The water flowing from Lake Vänern in Sweden is regulated by a barrage and by the output of the two units installed. These turbines are each rated at 11·2 MW at 46·9 r.p.m. under a design head of 4·3 m.

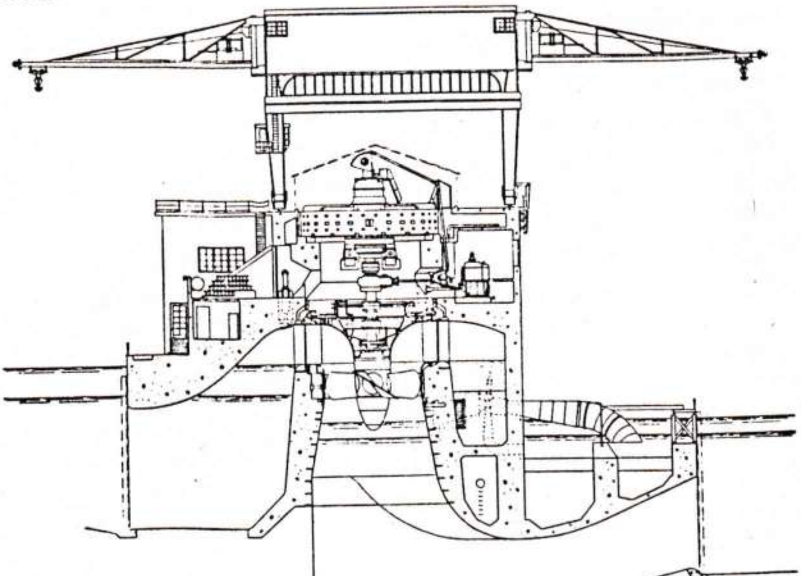

Fig. 4.26.—Vargön power station

The intake to the runner forms a siphon and thus eliminates the necessity of intake gates for the emergency shutting-down of the turbines. This feature, leading to an appreciable reduction in cost of the scheme, was obtained by adopting a runner setting such that the level of the turbine bottom cover was the same as the highest head-water level. A further economy was achieved by the "outdoor" power-house construction with removable covers for the generators.

With such an arrangement it is necessary to prime the turbine before it can be started by removing the air from the inlet casing. Ejectors are used for this purpose and governor-operated vacuum breakers are used for shutting down in an emergency.

The necessary choice of a specific speed which is lower than normal, and thus a lower value of sigma, resulted in a considerable reduction in excavation costs, which for a unit of this size is a significant advantage. Very careful final dressing of the lower rock surfaces led to a saving due to no concrete being required for lining parts of the draft tube.

It should be noted that the economic limit to such an arrangement precludes its use for heads much greater than 4·5 m. owing to the necessity for choosing a rather lower specific speed than normal.

15.2. *Muhammedpur Hydro-electric Scheme.*

The Muhammedpur power station (fig. 4.27) is located on the Ganges Irrigation Canal about 100 miles north-west of Delhi and houses three Kaplan turbines, each with an output of 3·16 MW. The power station is built adjacent to a bridge over the canal close to a former stilling pool, and the head is obtained by combining two falls in the canal which give a design head of 5·35 m. The headwater level is maintained constant over a wide range of flow in the canal by means of automatic tilting gates, each of which is located above the draft-tube exit of a turbine. The power station supplies power mainly for tube-well pumping in the vicinity.

The general arrangement is similar to many others located on canals where the drop in water level has been used to provide power according to the available flow. Each turbine is located in a separate channel and offsct a little in order to improve the flow of water entering the guide vanes. From fig. 4.27 it will be seen that the automatic tilting gates maintain the full head on the turbines during normal operation. If, however, the flow in the canal should be in excess of that required by the turbines, the level in the canal rises and causes the gates to open owing to the resulting increase in water pressure. The draft tubes have

a short axial length, so permitting the reduced excavation depth which was desirable owing to the nature of the alluvial subsoil and difficulties in dewatering the site during construction.

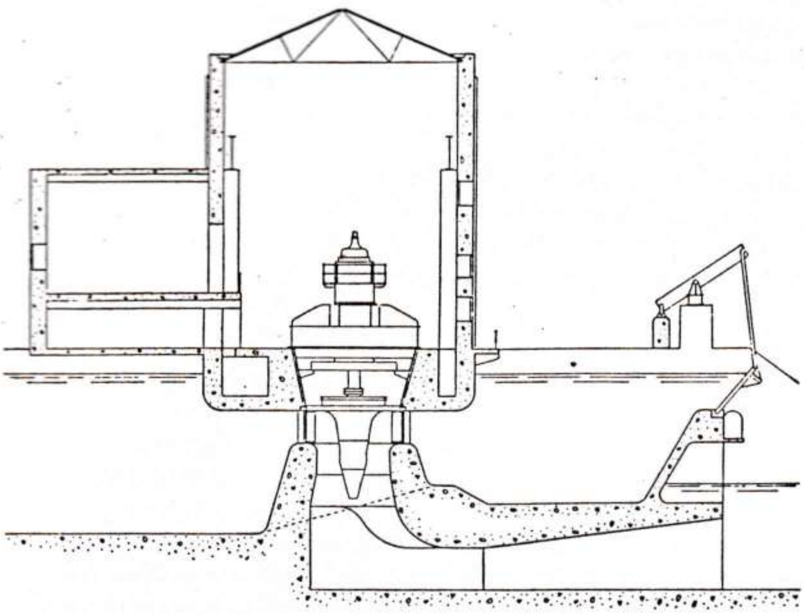

Fig. 4.27.—Muhammedpur power station

The turbines (fig. 4.3) are of the normal construction adopted for larger sizes operating under low heads, with outside regulation of the guide vanes. By alteration of the shape of the intake it has been possible to locate the servomotor at a level low enough to enable the connecting rod to be taken direct to the regulating ring instead of using the more expensive intermediate shaft, bearings, levers, etc., which are necessary when the servomotor is at a higher level. The hydraulic thrust, generator and turbine-runner weights are taken to the foundations by means of a fabricated structure. The grouted-in stays are also of fabricated-steel sections and of ample strength to carry these loads.

The water seals between the shafts and inner covers are of the labyrinth type as is usual when the tailwater level is below the runner centre-line. Any leakage water is removed either by electric motor-driven pumps or by the suction provided by a pipe leading from the top cover to the draft tube.

The bearings are of the oil-lubricated type with a revolving lower

oil-pot. Two Pitot tubes are provided for the circulation of oil, one being used with a small electric motor-driven pump so that the correct oil level in the upper reservoir can be established before starting up. Each unit is provided with a governor-oil pumping set with an electric motor-driven pump. The three pumping sets are interconnected so that in the event of failure of one of the pumps all of the three turbines can still be kept in operation.

Although propeller turbines—or perhaps two propeller turbines and one Kaplan turbine—could have been installed for this scheme, the former would have been appreciably larger and would have had either a lower speed or output. The ease of operation combined with the efficient use of the canal discharge with a somewhat higher overall efficiency over the working range has justified the slightly higher cost of the installation adopted.

15.3. *Karapiro Hydro-electric Scheme.*

The Karapiro hydro-electric scheme is one of a number of power developments, on the Waikato River in the North Island of New Zealand. Three Kaplan turbines are installed operating at a net head of 29·5–31·5 m., each unit being rated at 31·5 MW. at 166·7 r.p.m.

The turbines shown in fig. 4.7 were built in the early forties, and it is of interest to compare their design with that of the more modern Kindaruma machines (fig. 4.1) built in 1966. It will be seen that whereas grease-lubricated bearings were used on the Karapiro machines, the Kindaruma turbines have bearings of the self-lubricated type. Karapiro has a single guide-vane servomotor mounted on the concrete above the spiral casing, whilst the more modern Kindaruma turbine uses two smaller servomotors mounted on the top cover, thus simplifying the concrete shuttering round the turbine pit and dispensing with the regulating-ring roller race. The Kindaruma machine also utilizes the shaft oil-head system described on page 163 for operation of the runner-blade servomotor, whereas at Karapiro the combinator was mounted above the generator exciter, with the runner-blade regulating valve installed in the runner hub. The top cover of the Kindaruma machine is of fabricated construction, concentrically split to facilitate runner and shaft removal, whilst the older Karapiro turbine has a cast-iron cover split only radially.

At Karapiro each unit is equipped with a pumping set having one electric motor-driven oil-pump. The three pumping sets are interconnected. The actuator and pumping sets are located on the generator floor.

The decision to use the Kaplan in preference to the Francis type of turbine for this head is of some significance. The Francis turbine would probably have had a slightly higher peak efficiency than that obtained, which on test was slightly over 93 per cent, but only over a small range of output. Furthermore the Francis turbine would have been a little smaller but would have run at a lower speed, resulting in a rather higher generator cost. In initial capital cost, including the power-station structure, three Francis turbines would probably have been slightly cheaper. The capitalized value, however, of the extra revenue from three Kaplan turbines was considered to justify the extra cost. Further, the depth of excavation required to reach bedrock did not increase the capital cost to the disadvantage of the Kaplan turbine as it would have done had the bedrock level been higher.

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APPENDIX TO CHAPTER IV

16. BULB TURBINES

The progressive development of Kaplan and propeller turbines has facilitated the economic development of low-head hydro-electric schemes. However, the large size of these machines and the extensive associated civil works results in high costs per kilowatt of capacity or per kilowatt-hour of energy produced. As a result, hydro-power resources at very low heads cannot often be economically developed using conventional arrangements of Kaplan or propeller turbines.

The desire to develop very low-head schemes, of which tidal power is an example, has lead to the development of " bulb " turbines in which both turbine and alternator are mounted horizontally in the water passage and supported by fixed stay vanes.

The development of the bulb turbine passed through several stages. First attempts to reduce costs and improve efficiency produced an axial-flow turbine in a vertical, horizontal, or inclined water passage with the drive to the alternator taken through the water passage wall to the power station building; the two latter arrangements resulted in a rather wide building. This disadvantage was overcome by retaining

the horizontal layout, incorporating a bevel gearbox in the turbine hub and driving the alternator through a vertical shaft. The final step was to install the whole turbo-generator inside the water passage, thus virtually eliminating any superstructure in the form of a power station building. This evolution resulted in the true bulb turbine where the generator is encased in a plate-steel hydraulically-shaped bulb.

A different development is the "tubular" turbine in which the external rim of the turbine runner also carries the magnet ring of the generator, which runs in an annulus on the outside of the water passage. This arrangement was first patented by Harza in 1919 and 1924 and a number of such machines were built in Germany in 1938–1945. This arrangement is possible for fixed-blade runners.

The advantages of the bulb turbine over the conventional arrangement may be summarized as follows:

(1) Elimination of bends in the water passages.
(2) Reduction in length of flow passages.
(3) Use of multiple small machines built in the factory enables manufacturing costs to be reduced.
(4) Reduction in power station building costs is achieved—the machines can be incorporated within the dam or weir.
(5) Simplification and reduction of civil works, with consequent reductions in cost.

Bulb units can be classified broadly into two types. The first is a small-capacity unit using an asynchronous alternator, often running in oil. Simplicity and cheapness is the keynote, leading to factory assembly and fixed vanes on the runner.

For larger outputs, above about 4000 kW., a synchronous alternator running in air is adopted. If efficiency or high output under varying heads is important, a conventional Kaplan runner can be used. This arrangement enables the machine to act as a pump or generator with either forward or reverse flow and has been adopted for the Rance tidal-power project.

The above outline of bulb turbines can be supplemented by reference to the following bibliography.

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